federal register

Thursday
April 19, 1979

Part II

Department of Energy

Office of Energy Conservation and Solar Applications

Proposed Rulemaking and Public Hearing Regarding Test Procedures for Central Air Conditioners, Including Heat Pumps

DEPARTMENT OF ENERGY

Office of Energy Conservation and Solar Applications

[10 CFR Part 430]

Proposed Rulemaking and Public Hearing Regarding Test Procedures for Central Air Conditioners, Including Heat Pumps

AGENCY: Department of Energy.

ACTION: Proposed rule.

SUMMARY: The Department of Energy hereby proposes to amend its test procedures for central air conditioners to include test procedures for heat pumps. These test procedures are part of the energy conservation program for consumer products established pursuant to the Energy Policy and Conservation Act. Among other program elements, the legislation requires that standard methods of testing be prescribed for covered products.

DATES: Comments by June 15, 1979; requests to speak by June 1, 1979; statements by June 8, 1979; hearing to be held on June 13, 1979.

ADDRESSES: Comments and requests to speak at the hearing to: Margaret W. Sibley, Conservation and Solar Applications, Department of Energy, Room 2221C, 20 Massachusetts Avenue, N.W., Washington, D.C. 20545.

Hearing held at: Department of Energy, Room 2105, 2000 M Street, N.W., Washington, D.C. 20461.

FOR FURTHER INFORMATION CONTACT:

James A. Smith, U.S. Department of Energy, Office of Conservation and Solar Applications, Division of Buildings and Community Systems, Consumer Products Efficiency Branch, 20 Massachusetts Avenue, N.W., Washington, D.C. 20545, (202) 376–4814.

Margaret W. Sibley, U.S. Department of Energy, Office of Conservation and Solar Applications, 20 Massachusetts Avenue, N.W., Washington, D.C. 20545, (202) 376–1651.

William J. Dennison, Office of the General Counsel, 20 Massachusetts Avenue, N.W., Room 3228, Washington, D.C. 20545, (202) 376–4100.

SUPPLEMENTARY INFORMATION:

A. Background

On October 1, 1977, the Department of Energy (DOE) assumed the authority of the Federal Energy Administration (FEA) for the energy conservation program for consumer products, pursuant to Section 301 of the Department of Energy Organization Act (DOE Act) (Pub. L. 95–91). The energy conservation program for consumer products was established by FEA pursuant to Title III, Part B of the Energy Policy and Conservation Act (Act) (Pub. L. 94–163). Subsequently, the Act was amended by the National Energy Conservation Policy Act (NECPA) (Pub. L. 95–619). References in this notice to "the Act" or to sections of the Act refer to the Energy Policy and Conservation Act as amended by NECPA. Section 323 (42 U.S.C. 6293) of the Act requires that standard methods of testing be prescribed for covered products. Test procedures appear at 10 CFR Part 430, Subpart B.

Test procedures were proposed for central air conditioners by notice issued June 7, 1977 (42 FR 30401, June 14, 1977) and a public hearing on the proposed test procedures was held on August 4, 1977. Final test procedures for central air conditioners were prescribed on November 21, 1977 (42 FR 60150, November 25, 1977).

DOE is today proposing to amend the test procedures for central air conditioners to include testing procedures for heat pumps.

B. Discussion

The heat pump is basically a variety of air conditioner which utilizes the mechanical refrigeration cycle to provide heating and may also provide cooling. DOE is proposing to include heat pumps as a class of central air conditioners because the primary function(s) of the equipment are essentially identical.

The definition of central air conditioners is proposed to be modified to include both heating and cooling units. In order to clarify terms, definitions of "heat pump", "air-source heat pump", "water-source heat pump", and "cooling only unit", are proposed to be added in Section 430.2. Other new terms which specifically apply to the laboratory methodology for heating performance are proposed to be included in Section 1 of the proposed Appendix M.

The test methodology for cooling only units remains the same as previously adopted for central air conditioners, but the organization of the test procedure as it appears in the Code of Federal Regulations has been revised so the heating performance methodology can be incorporated and the total procedure can be more understandable for the user. Sections 1, 2.1, 3.1, 4.1, and 5.1 of proposed Appendix M contain the test procedures for central air conditioners as previously developed, while all other sections refer to new proposals concerning heat pumps. The only signficant change to the existing central air conditioner test procedure is the inclusion of the cooling seasonal performance factor (CSPF) in order to maintain continuity between the proposed measure of heating efficiency and the measure of cooling efficiency. With this change, the measures may be easily combined or weighted to give an annual efficiency which may be used by consumers in purchasing decisions. DOE expects that the measure of efficiency used in the energy standards program for central air conditioners will be based on the cooling seasonal performance factor (CSPF), the heating seasonal performance factor (HSPF), or the annual performance factor (APF). These terms are defined in sections 1.1, 1.9, and 1.15 of the proposed Appendix M. Although the major additions have to do only with testing for heating performance, interested parties may comment on the cooling as well as the heating portion.

Air-source heat pumps are equipped with a means of providing supplementary heat by means of electrical resistance or a fossil fuel furnace. These systems are required at low outdoor temperatures when the heat pump itself cannot provide the required heating to satisfy the building load. It should be understood that the proposed rule as it applies to air-source heat pumps only covers those units which are designed for use with electrical resistance back up systems. DOE is currently developing test procedures for other types of back up systems, including gas and oil furnaces, and intends to propose these as soon as practicable.

As described in the definition of "central air conditioner" only units which are electrically powered are covered by these procedures. Available information indicates that gas-fired heat pumps and gas-fired central conditioners are not distributed in commerce to any significant extent and therefore are not included in this proposal.

Section 323(a)(2) of the Act requires DOE to direct the National Bureau of Standards (NBS) to develop test procedures for the determination of the estimated annual operating costs and at least one other useful measure of energy consumption which DOE determines is likely to assist consumers in making purchasing decisions. At DOE's request NBS evaluated existing test procedures for measuring energy consumption of heat pumps. Present industry testing

procedures for heat pumps use only steady state conditions and do not account for performance degradation due to defrosting and cycling of the unit.

NBS transmitted to DOE a test procedure review document which reviewed existing test procedures for measuring the energy consumption of heat pumps and which recommended a test procedure. Copies of this review document will be made available for inspection by interested persons as provided for later in this notice.

The test procedure amendments recommended by NBS and proposed today incorporate, in part, test procedures used by industry to measure the heating capacity and efficiency of heat pump equipment as described in the Air Conditioning and Refrigeration Institute (ARI) Standards 240–77, 210–75, and 320–76 and the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) Standard 37–69. In addition, the amendments incorporate procedures for determining estimated annual operating costs and seasonal performance factors of heat pumps in both the heating and cooling modes.

Pursuant to section 301 of the DOE Act, DOE is required to comply with section 32 of the Federal Energy Administration Act of 1974 (FEA Act) (Pub. L. 93–275), as amended by section 9 of the Federal Energy Administration Authorization Act of 1977 (Pub. L. 95–70).

Sections 32(a) and (b) of the FEA Act provide that:

(a) If any proposed rule by the Administrator contains any commercial standards, or specifically authorizes or requires the use of any such standards, then any general notice of the proposed rulemaking shall:

(1) identify, by name, the organization which promulgated such standards; and

(2) state whether or not, in the judgment of the Administrator, such organization complied with the requirements of subsection (b) in the promulgation of such standards.

(b) An organization complies with the requirements of this subsection in promulgating any commercial standards if:

(1) it gives interested persons adequate notice of the proposed promulgation of the standards and an opportunity to participate in the promulgation process through the presentation of their views in hearings or meetings which are open to the public;

(2) the membership of the organization at the time of the promulgation of the standards is sufficiently balanced so as to allow for the effective representation of all interested persons;

(3) before promulgating such standards, it makes available to the public any records or proceedings of the organization, and any documents, letters, memorandums, and materials relating to such standards; and

(4) it has procedures allowing interested persons to:

(A) obtain a reconsideration of any action taken by the organization relating to the promulgation of such standards, and

(B) obtain a review of the standards (including a review of the basis or adequacy of such standards).

The findings required of DOE by section 32 of the FEA Act serve to alert the public and DOE to the use and background of commercial standards in a proposed rulemaking and, through the comment and hearing process, allow interested persons to make known their views regarding the appropriateness of the use of those particular commercial standards in that proposed rulemaking. Any negative finding does not reflect a determination by DOE of the advisability of utilizing a particular standard for part of the test procedure regulation.

As noted above, the rulemaking proposed today contains elements of four commercial standards promulgated by two non-governmental organizations, ASHRAE and ARI.

Determinations have not been made regarding ASHRAE because the ASHRAE standard cited is part of the existing central air conditioner test procedure and was previously subjected to review under section 32(c) of the FEA Act. With respect to the ARI standards cited, DOE has determined that ARI does not comply with the requirements of section 32(b) because among other things, its membership is limited to manufacturers of air conditioning and refrigeration equipment and, therefore is not "sufficiently balanced so as to allow for the effective representation of all interested persons." As required by Section 32(c), DOE will consult with the Attorney General and the Chairman of the Federal Trade Commission concerning the impact of these standards on competition, prior to prescribing final test procedures.

C. Measures of Energy Consumption

The Act requires DOE to prescribe test procedures for the determination of estimated annual operating costs and at least one other useful measure of energy consumption which the Secretary determines is likely to assist consumers in making purchasing decisions.

DOE is proposing to amend § 430.22 by substituting a new paragraph (m) to include all measures of energy consumption specified for all classes of central air conditioners including heat pumps. The following is a brief description of each measure.

The estimated annual operating cost for the cooling and/or heating functions of central air conditioners is an estimate of the electrical energy input necessary to satisfy a representative annual heating and/or cooling load multiplied by a representative average unit cost of electricity. For air-source central air conditioners (section 430.22(m)(1)), the annual electrical energy input for cooling is estimated by dividing the output (the cooling capacity times the representative annual average cooling load hours) by the seasonal energy efficiency ratio. The annual electrical energy input for heating is estimated by dividing the output (a representative design heating requirement times a representative average heating load hours) by a representative heating seasonal performance factor. For water-source heat pumps (§ 430.22(m)(2)), the annual electrical energy input is estimated by multiplying the test measured electrical power input by the representative annual average cooling or heating load hours.

Also proposed (§ 430.22(m) (5) and (6)) as test procedures for the determination of other measures of energy consumption which are likely to assist consumers in making purchasing decisions are test procedures regarding the measures of efficiency: The seasonal energy efficiency ratio, as described in 430.22(m)(5)(iv), the cooling seasonal performance factor, as described in 430.22(m)(5)(i) and 430.22(m)(6)(i), the heating seasonal preformance factor, as described in 430.22(m)(5)(ii) and 430.22(m)(6)(ii), and the annual performance factor, as described in 430.22(m)(5)(iii) and 430.22(m)(6)(iii). The cooling seasonal performance factor (CSPF) and the seasonal energy efficiency ratio (SEER) are an estimate of the total cooling output during the normal annual usage period for cooling, divided by an estimate of the total electric input during the same period. The heating seasonal performance factor (HSPF) is an estimate of the total heating output during the normal annual usage period for heating, divided by an estimate of the total electric input during the same period. The annual performance factor (APF) is a weighted average of the combined HSPF and CSPF depending on the number of annual heating load hours and cooling

load hours for a specific region of the country.

Additional proposed measures which are likely to assist consumers in making purchasing decisions are the estimated regional annual operating costs for central air conditioners (§ 430.22(m) (3) and (4)). The estimated regional annual operating cost for the cooling and/or heating functions of central air conditioners is an estimate of the annual electrical energy input necessary to satisfy a local heating and/or cooling load multiplied by a representative average unit cost of electricity. The annual electrical energy input is estimated in the same manner as detailed above except that regional annual cooling and heating load hours (from sections 6.1.3. and 6.2.5 in Appendix M to Subpart B) are performance factors described in 430.22(m) (5) and (6) are used.

DOE recognizes that there may be additional useful measures of energy consumption for central air conditioners other than the measures described above. Accordingly, today's proposed amendment in paragraph 430.22(m)(7) provides for other useful measures which the Secretary determines are likely to assist consumers in making purchasing decisions. These measures, however, must be derived from the application of the uniform test method proposed today as Appendix M to Subpart B.

D. Laboratory Methodology

The proposed rule provides for a controlled laboratory environment for measuring energy consumption of all central air conditioners, including heat pumps. The test methodology for the air-source cooling mode is the same as adopted in the existing central air conditioner test procedure. Appendix M has been renumbered with the air-source cooling protion of the test methodology appearing in sections 2.1, 3.1, 4.1 and 5.1. Air-source heating test methodology appears in sections 2.2, 3.2, 4.2, and 5.2. Water-source heating and cooling methodologies appear in sections 2.4, 3.4, 4.4, 5.4, and 5.5. All reference material is now incorporated in sections 6.1 through 6.4.

The proposed amendments specify methodologies for measuring the heating performance of both air-source and water-source heat pumps. The test method for air-source units in the heating mode utilizes four tests to evaluate: the steady state efficiency, the energy lost to defrosting the outdoor coil, the degradation effect caused by cycling of the unit, and the efficiency at low outdoor temperatures. Utilizing the results of these tests, an efficiency curve can be generated which is applicable to all outdoor temperatures that may occur over the course of a heating season. A mathematical summation technique is then used incorporating the weather data of different regions across the country to calculate the perfomance factors for a given unit in a particular location. Because an air-source heat pump has different performance depending on outdoor temperature, a given unit will have varying performance depending on the geopraphical location of the unit.

In order to reduce the burden of increased testing on manufacturers with limited capability for testing heating performance, an assigned degradation coefficient, C_D, may be used in lieu of condicting the Cyclic Test on air-source heat pumps. If this option is chosen the value substituted is proposed to be 0.25 in all cases. The derivation of this factor is consistent with the present methodology used in the cooling performance testing. If a manufacturer chooses this option, he must then use the assumed value in calculations for all his basic models. Another technique used to reduce testing is the use of a factor to show improved seasonal performance on units with sophisticated "demand-defrost control systems". Due to the complexity of a test that would be required to show the actual difference in seasonal performance between demand-defrost and the conventional time-temperature defrost systems, an estimated value of 5 percent improvement in the defrost capacity was assigned to units which utilize demand-defrost control systems. This factor may be adjusted by DOE depending on data presented during the comment period.

E. Representative Average-Use Cycle

Section 323(b)(2) (42 U.S.C. 6293(b)(2)) of the Act provides that test procedures for determining estimated annual operating costs of any covered product shall be calculated from measurement of energy use in a representative average-use cycle and from representative average unit costs needed to operate such product during such cycle. DOE proposes that the original representative average-use cycle of 1000 cooling load hours per year for the cooling mode be applicable for both cooling only and heating/cooling units. DOE has also determined that 2080 heating load hours is the representative average-use cycle for central air conditioners in the heating mode. These determinations are based upon NBS' recommendation, which in turn is based upon NBS' reports which estimated the annual compressor operating hours for central air conditioners in the cooling mode and the annual heating load hours for furnances. Because the average number of heating load hours is dependent on average weather data and the number of units in a given area, it is applicable for all types of heating equipment.

F. Unit Costs of Energy

Under section 323(b)(2) of the Act, DOE is required to provide manufacturers with information on representative average unit costs of energy. This information was provided by notice issued July 11, 1977 (42 FR 36549, July 15, 1977) and is expected to be updated on a continuing basis.

G. Number of Units to be Tested

On September 8, 1978, (43 FR 40192) DOE proposed to amend test procedures for all types of consumer products including central air conditioners, currently covered by this program, in order to prescribe sampling provisions applicable to any testing required under section 323(c) or 324 of the Act. This testing related specifically to representations regarding the energy consumption of or cost of energy consumed by the various products, and to the labeling of these products with energy consumption of or cost of energy consumed by the various products, and to the labeling of these products with energy consumption information as required by the Federal Trade Commission (FTC).

DOE is today reproposing § 430.22(m) to receive comments on the sampling approach to be adopted for central air conditioners, including heat pumps. The major proposed changes from the present sampling requirements include the use of one sided sampling provisions in lieu of two sided, the use of computer modeling mathematical correlation to rate the condenser-evaporator-coil combinations other than those actually tested, and the ability to substitute certain components within a system if measures of energy consumption are not affected.

Those commenting on the September 8, 1978, notice recommended that DOE adopt sampling plans which required fewer units to be tested as compared to either of the distinct plans presented in DOE's proposed sampling rule or FTC's proposed labeling rule which appeared in the **Federal Register** on July 21, 1978 (43 FR 31806).

The two rules proposed by DOE and FTC would have had different effects on the extent of testing. The rule proposed by FTC specified that units of every basic model need be tested. The rule

proposed by DOE contained provisions which had the effect of specifying that units of only certain basic models need be tested. Specifically, the DOE rule grouped basic models according to the condensing unit used and required that only units of the condenser-evaporator-coil-combination (defined in DOE test procedures) which was expected to have the largest volume of retail sales among all basic models using that condensing unit need be tested. Both the DOE and FTC proposed rules specified 90 percent confidence and tolerances of ±5% for their sampling plans, and both utilized two-sided confidence limits.

Commenters requested that any sampling plan adopted should require the testing of only certain basic models, and for those basic models tested, the statistical parameters should be modified to require fewer units to be tested as compared to either proposed rule.

Commenters provided information on the time required to test one unit; the frequency of introducing new basic models; the number of units of each basic model which would have to be tested to satisfy sampling provisions using modified sampling parameters; the extent that existing test results can be used to fulfill the requirements of Federal program; the extent of testing beyond the level previously conducted to support programs either administered by trade associations or conducted for the manufacturers' own purposes; the number of basic models the commenter currently produces; the energy consuming characteristics or features that distinguish the commenter's basic models; the capacity of test facilities used by the commenter; and the cost of testing all the commenter's basic models.

Commenters urged that if units of every basic model were to be tested, testing costs could be expected to be $65 million. DOE has reviewed the data submitted and methods by which this figure was determined, and believes that actual costs would be substantially lower, but may still be significant.

Commenters also stated that measures of energy consumption of basic models not tested could be determined by engineering analysis based on data obtained by testing other basic models which used the same condensing unit. It was argued that this method of determining measures of energy consumption would be accompanied by very little sacrifice in reliability of such measures. The DOE proposed sampling rule permitted the determination of measures of energy consumption in this manner. Industry-wide, the number of basic models which would require actual testing would be thus reduced by a factor of about 5, with a commensurate reduction in the costs of testing. DOE is proposing that among all basic models using a single condensing unit, only units of the condenser-evaporator-coil combination expected to have the largest volume of retail sales need be tested.

Other comments indicated that DOE's proposed sampling plan was logically inconsistent in that it specified confidence levels and tolerances for data not supported by testing. DOE agrees that any simulation employing computer programs or engineering analysis to determine measures of energy consumption cannot reasonable demonstrate that the true mean of such measures is contained in a certain range about represented values. Therefore today's proposed rules does not specify such confidence levels and tolerances for data not supported by testing. In order to provide manufacturers a guideline for determining measures of energy consumption applicable to all the units of all basic models. The proposed rule describes certain minimum criteria for computer simulations which may be used in place of testing certain basic models. Whenever measures of energy consumption determined by computer simulation do not agree with measures of energy consumption determined by actual testing, the values determined by testing will be assumed to be the more reliable values.

The basis of the simulation technique must be a computer model of the mechanical vapor-compression refrigeration cycle. The major components in the refrigeration cycle are modeled as "fits" to manufacturer component performance data. For example, heat transfer characteristics of the coils are modeled as functions of face area, number of rows, air flow rate, fins per inch, and entering air enthalpy. Compressor performance curves must be used to provide refrigerant mass flow rate and compressor power as a function of suction temperature and condenser temperature. An iterative technique must then be used which varies the evaporator temperature, condensing temperature, and refrigerant mass flow rate to balance the cycle. When the cycle is balanced, the enthalpies of the process end points are knows as well as the refrigerant mass flow rate, and, as a result, the measure of steady state efficiency can be calculated.

The computer model is then to be calibrated to actual test results and may be used to predict the energy efficieny of basic models which use the same condensing unit as a condenser-evaporator-coil combination which has been tested. In order to insure the accuracy of ratings derived in this manner, and to insure compatibility between manufacturers, any manufacturer which uses the computer modeling provisions is required to submit to DOE a description of the analysis used, including the computer model and the data and analysis used to verify the accuracy of the ratings obtained by computer modeling. At this time DOE does not plan to treat such information as confidential or proprietary, but is interested in receiving comments on this issue. DOE is also interested in receiving comments on the feasibility of DOE prescribing a computer model for use by all manufacturers who elect to rate their products in this manner.

Commenters stated that it is often necessary to substitute components manufactured by different suppliers and that these substitutions typically have a minimal effect on energy consumption. Commenters further stated that based on certain of the definitions of the term "basic model" presented in DOE test procedures, it was not clear whether the units which include such component substitutions constituted a new basic model which would require additional testing.

If the tested values of the measures of energy consumption for units having such component substitutions change by only a minimal amount from the tested values before the substitution, DOE agrees that additional testing of such units would not provide benefit to the consumer commensurate with costs incurred. For example, if a 350 watt condenser fan in a central conditioner is substituted for a 330 watt condenser fan and if this is the only design change which affects the tested values of the measures of energy consumption, then the total average power needed to operate the unit may be expected to increase by 20 watts. However, as compared to the total average power consumption of 4350 watts, any changes in the tested values of the measures of energy consumption will be less than errors caused by the numerical rounding specified in DOE test procedures. Therefore, a sample of units containing the substituted condenser fan need not be tested since, by not changing the values of the tested measure of energy consumption, the confidence statement remains satisfied for the represented values of the measures of energy consumption. However, if a 3500 watt compressor motor in a central air

conditioner is substituted for a 3300 watt compressor motor of identical capacity, then the efficiency of such units is likely to decrease and their energy consumption is likely to increase. It is probable that the represented value of the measure of efficiency based on testing units containing the 3300 watt compressor motor will no longer satisfy the sampling confidence statement. In such case, further testing would be needed to establish the represented value of the measures of energy consumption which will satisfy the confidence statement for units which use the 3500 watt compressor motor

To clarify the test procedures, wherever the term "basic model" is used in § 430.23, a footnote states that components of similar design may be substituted without necessitating further testing, if the represented values of the measures of energy consumption will not change as a result of the substitution, i.e., the represented values continue to satisfy the sampling confidence statement. With the adoption of sampling plans utilizing one-sided confidence limits, manufacturers may adjust the represented values of the measures of energy consumption from tested values of the measures of energy consumption to compensate for anticipated changes in the energy consuming characteristics of future production units that may include such component substitutions.

After reviewing the data submitted, DOE by this notice is reproposing a sampling plan which specifies 90 percent confidence and 5 percent tolerances on a one-sided sampling system. DOE believes that this system provides the most reliable data for determining measures of energy consumption when compared against increased costs for testing.

An alternative approach under consideration by DOE is a plan in which manufacturers would be required to test two randomly selected units of the condenser-evaporator-coil combination likey to have the largest volume of sales. The measure of energy consumption used in labeling and representations would be required to be no higher than the mean of the values of efficiency derived from testing two units or no lower than the mean of the values of operation cost derived from the same testing. The advantage of this method would be to limit required testing to no more than two units.

Computer modeling would then be used for other basic models which use the same condensing unit. Manufacturers might still rate conservatively depending on their prior knowledge of the coefficient of variation of the particular product.

A disadvantage of this method is that the reliability of the measures of energy consumption could be expected to be high for basic models where test results among the units tested show little variability, and the reliability could be expected to be low where there was substantial variability. It should be noted that a major purpose for testing is to provide reliable measures of energy consumption in order to assist consumers in making purchasing decisions.

Manufacturers and other interested persons are encouraged to comment on the proposed approach and the alternative. All comments and data pertaining to the alternate approach should be detailed specifically to provide a basis for incorporation into the final rule, in the event that the alternate approach is chosen for the final rule.

H. Comment Procedure

1. *Written Comment.* Interested persons are invited to participate in this rulemaking by submitting data, views or arguments with respect to the proposed amendments set forth in this notice.

Comments should be identified on the outside of the envelope and on documents submitted to DOE with the designation "Central Air Conditioners—Proposed Amendments (Docket No. CAS–RM–79–102)." Fifteen copies should be submitted. All comments received by June 15, 1979, and all other relevant information, will be considered by DOE before final action is taken on the proposed regulation.

Pursuant to the provisions of 10 CFR 1004.11(e), any person submitting information which he believes to be confidential should so identify the information and submit one copy only of the information. DOE reserves the right to determine the confidential status of the information or data and treat it according to its determination.

2. *Public Hearing.* a. *Request Procedure*—The time and place of the public hearing are indicated at the beginning of this preamble. The hearing will be continued, if necessary on June 14, 1979. DOE invites any person who has an interest in the proposed amendment issued today, or who is a representative of a group or class of persons that has an interest in today's proposed amendment, to make a written request for an opportunity to make an oral presentation. Such a request should be directed to the address indicated at the beginning of this preamble and must be received before June 1, 1979. Such a request may be hand delivered to such address, between the hours of 8 a.m. and 4:30 p.m., Monday through Friday. A request should be labeled both on the document and on the envelope "Central Air Conditioners—Proposed Amendments (Docket No. CAS–RM–79–102)."

The person making the request should briefly describe the interest concerned; if appropriate, state why she or he is a proper representative of a group or class of persons that has such an interest, and given a concise summary of the proposed oral presentation and a telephone number where she or he may be contacted through.

DOE will notify, before 4:30 p.m., June 4, 1979, each person selected to appear at the hearing. Each person selected to be heard must submit 15 copies of her or his statement to the address and by the date given in the beginning of this preamble. In the event any person wishing to testify cannot meet the 15 copy requirement, alternative arrangements can be made with Margaret W. Sibley in advance of the hearing by so indicating in the letter requesting an oral presentation or by calling (202) 376–1651.

b. *Conduct of Hearing.* DOE reserves the right to select the persons to be heard at the hearing, to schedule their respective presentations and to establish the procedures governing the conduct of the hearing. The length of each presentation may be limited, based on the number of persons requesting to be heard.

A DOE official will be designated to preside at the hearing. This will not be a judicial or evidentiary type hearing. Questions may be asked only by those conducting the hearing, and there will be no cross-examination of persons presenting statements. Any decision made by DOE with respect to the subject matter of the hearing will be based on all information available to DOE. At the conclusion of all initial oral statements, each person who has made an oral statement will be given the opportunity if she or he so desires, to make a rebuttal statement. The rebuttal statements will be given in the order in which the initial statements were made and will be subject to time limitations.

Any interested person may submit questions to be asked of any person making a statement at the hearing to Margaret W. Sibley, Conservation and Solar Applications, DOE, 20 Massachusetts Avenue, NW., Washington, D.C. before 4:30 p.m. June 8, 1979. DOE will determine whether the question is relevant, and whether the

time limitations permit it to be presented for answer.

Any further procedural rules needed for the proper conduct of the hearing will be announced by the presiding officer.

A transcript of the hearing will be made and the entire record of the hearing, including the transcript, will be retained by DOE and made available for inspection at the DOE Freedom of Information Office in the Forrestal Building, Independence Avenue and L'Enfant Plaza, SW., Washington, D.C. 20585, between the hours of 8 a.m. and 4:30 p.m., Monday through Friday. In addition, any person may purchase a copy of this transcript from the reporter.

I. Environmental and Inflationary Review

Pursuant to Section 7(c)(2) of the Federal Administration Act of 1974, a copy of this notice has been submitted to the Administrator of the Environmental Protection Agency for his comments concerning the impact of this proposal on the quality of the environment. The Administrator has no comments at the present time.

The National Environmental Policy Act of 1969 requires DOE to assess the environmental impacts of any proposal issued by the Department for "major Federal actions significantly affecting the quality of the human environment." Since test procedures under the energy conservation program for consumer products will be used only to standardize the measurement of energy usage and will not affect the quality or distribution of energy usage, DOE has determined that the action of prescribing test procedures, by itself, will not result in any environmental impacts. On this basis, DOE has determined that, with respect to prescribing test procedures under the energy conservation program for consumer products, no environmental impact statement is required.

The proposed rule has been reviewed in accordance with Executive Order 12044 and DOE Order 2030. From the regulatory analysis performed, it was determined that the proposal was significant in nature but did not have major impacts to manufacturers and consumers (imposing annual economic costs of $100 million or more).

(Energy Policy and Conservation Act, Pub. L. 94–163, as amended by Pub. L. 94–619; Department of Energy Organization Act, Pub. L. 95–91; E.O. 12009, 42 FR 46267.)

In consideration of the foregoing, it is proposed to amend Chapter II of title 10, Code of Federal Regulations, as set forth below.

Issued in Washington, D.C., April 6, 1979.

Omi G. Walden,

Assistant Secretary, Conservation and Solar Applications.

1. Section 430.2 is amended by deleting the present definition of "central air conditioner" and "condensing unit" and adding the following new definitions inappropriate alphabetical order:

§ 430.2 Definitions.

* * * * *

"Air-source heat pump" means a "heat pump" which consists of one or more assemblies, which utilizes an indoor conditioning coil, compressor(s), and refrigerant-to-outdoor-air heat exchanger to provide air heating, and which may also provide air cooling, dehumidifying, circulating, and air cleaning.

* * * * *

"Central air conditioner" means a consumer product which is powered by single phase electric current, which is rated below 65,000 Btu/hour, and which is either a "heat pump" or a "cooling only unit".

* * * * *

"Condensing unit" means a component of a central air conditioner which is designed to remove the heat absorbed by the refrigerant and to transfer it to the outside environment, and which consists of an outdoor coil, compressor(s), and air moving device.

* * * * *

"Cooling only unit" means a "central air conditioner" which consists of a compressor(s), an air cooled condensing unit, and an evaporator or cooling coil, which is designed to provide air cooling, dehumidifying, circulating, and air cleaning.

* * * * *

"Heat pump" means a "central air conditioner" which is either an "air-source heat pump" or a "water-source heat pump".

* * * * *

"Water-source heat pump" means a "heat pump" which consists of one assembly which utilizes an indoor conditioning coil with air moving means, compressor(s), and refrigerant-to-water heat exchanger(s) to provide both air heating and cooling, dehumidifying, circulating, and air cleaning.

* * * * *

2. Section 430.22 is amended by substituting a new paragraph (m) to read as follows:

§ 430.22 Test procedures for measures of energy consumption.

* * * * *

(m) Central Air Conditioners. (1) The estimated annual operating cost for cooling only units and air-source heat pumps shall be one of the following:

(i) For cooling only units or the cooling portion of the estimated annual operating cost for air-source heat pumps which provide both heating and cooling, the product of (A) the quotient of the cooling capacity, in Btu's per hour, determined from the steady-state wet-coil test (test A) measured at the highest compressor speed, as described in Section 3.1 of Appendix M to this subpart, and the average seasonal energy efficiency ratio, in Btu's per watt-hour, determined from section 5.1 of Appendix M to this subpart; (B) the representative average use cycle of 1000 cooling load hours per year; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year;

(ii) For heat pumps which provide only heating or the heating portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, the product of (A) the quotient of the standardized design heating requirement nearest the capacity measured in the High Temperature Test, $Q_{ss}(47)$ in Btu's per hour, determined in sections 5.2 and 6.2.6 of Appendix M to this subpart, and the heating seasonal performance factor corresponding to the above mentioned standardized design heating requirement determined from section 5.2 of Appendix M to this subpart; (B) the representative average use cycle of 2080 heating load hours per year; (C) a correction factor described in section 5.2 of Appendix M to this subpart; (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act; and (E) a conversion factor of one kilowatt-hour per 3,413 Btu's, the resulting quotient then being rounded off to the nearest dollar per year; or

(iii) For central air conditioners which provide both heating and cooling, the quantity determined in paragraph (i) above added to the quantity determined in paragraph (m)(1)(ii) above.

(2) The estimated annual operating cost for water-source heat pumps shall be the sum of:

(i) The product of (A) the electrical power input in watts, and described by the quantity E_T in section 5.4 of

Appendix M to this subpart; (B) the representative average use cycle of 1,000 cooling load hours per year; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year, and

(ii) The product of (A) the electrical power input in watts, as described by the quantity E_H in section 5.5 of Appendix M to this subpart; (B) the representative average use cycle of 2080 heating load hours per year; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(a) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(3) The estimated regional annual operating cost(s) for cooling only units and air-source heat pumps shall be one of the following:

(i) For cooling only units or the cooling portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, the product of (A) the quotient of cooling capacity in Btu's per hour determined from the steady-state wet-coil test (test A) measured at the highest compressor speed, and the average seasonal energy efficiency ratio in Btu's per watt-hour both determined from section 5.1 of Appendix M to this subpart; (B) the estimated number of local cooling load hours per year determined from section 6.1.3 of Appendix M to this subpart; (C) a conversion factor of .001 kilowatts per watt; and (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year;

(ii) For heat pumps which provide only heating or the heating portion of the estimated annual operating cost for central air conditioners which provide both heating and cooling, for each applicable standardized design heating requirement, the product of (A) the quotient of the estimated number of local heating load hours per year determined from section 6.2.5 of Appendix M to this subpart, and the heating seasonal performance factor corresponding to the appropriate standardized design heating requirement determined from section 5.2 of Appendix M to this subpart; (B) 0.77, a correction factor described in section 5.2 of Appendix M to this subpart; (C) the standardized design heating requirement in Btu's per hour determined in sections 5.2 and 6.2.6 of Appendix M to this subpart; (D) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act; and (E) a conversion factor of one kilowatt-hour per 3,413 Btu's, the resulting quotient then being rounded off to the nearest dollar per year; or

(iii) For central air conditioners which provide both heating and cooling, the quantity determined in paragraph (m)(3)(i) above added to each quantity determined in paragraph (m)(3)(ii) above.

(4) The estimated regional annual operating cost for water-source heat pumps shall be the sum of:

(i) The product of (A) the quotient of the cooling capacity from test F in Btu's per hour and the cooling seasonal performance factor as determined in section 5.4 of Appendix M to this subpart for the local ground water temperature determined from section 5.4 and 6.4 of Appendix M to this subpart; (B) the estimated number of local cooling load hours per year determined from section 6.1.3 of Appendix M to this subpart; (C) a conversion factor of .293 watt-hours per Btu; (D) a conversion factor of .001 kilowatts per watt; and (E) a representative average unit cost of electricity in dollars per kilowatt hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar year, and

(ii) The product of (A) the quotient of the heating capacity from test G, in Btu's per hour, and the heating seasonal performance factor as determined in section 5.5 of Appendix M to the subpart for the local ground water temperature determined from section 5.5 and 6.4 of Appendix M to this subpart; (B) the estimated number of local heating load hours per year determined from section 6.2.5 of Appendix M to this subpart; (C) a conversion factor of .293 watt-hours per Btu: (D) a conversion factor of .001 kilowatts per watt; and (E) a representative average unit cost of electricity in dollars per kilowatt-hour as provided pursuant to section 323(b)(2) of the Act, the resulting product then being rounded off to the nearest dollar per year.

(5) The measure(s) of efficiency for cooling only units and air-source heat pumps shall be one or more of the following:

(i) The cooling seasonal performance factor for cooling only units and air-source heat pumps which provide cooling shall be the cooling seasonal performance factor determined according to section 5.1 of Appendix M to this subpart, rounded off to the nearest 0.01.

(ii) The heating seasonal performance factors for air-source heat pumps shall be the heating seasonal performance factors determined according to section 5.2 of Appendix M to this subpart for each applicable standardized design heating requirement within each climatic region, rounded off to the nearest 0.01.

(iii) The annual performance factors for air source heat pumps which provide heating and cooling, shall be the annual performance factors determined according to section 5.3.1 of Appendix M to this subpart for each standardized design heating requirement within each climatic region, rounded off to the nearest 0.01.

(iv) The seasonal energy efficiency ratio for cooling only units and air-source heat pumps which provide cooling shall be the seasonal energy efficiency factor determined according to section 5.1 of Appendix M to this Subpart, rounded off to the nearest 0.01.

(6) The measure(s) of efficiency for water-source heat pumps shall be one or more of the following: (i) The cooling seasonal performance factors for water-source heat pumps shall be the cooling seasonal performance factors determined according to section 5.4 of Appendix M to this subpart, rounded off to the nearest 0.01.

(ii) The heating seasonal performance factors for water-source heat pumps shall be the heating seasonal performance factors determined according to section 5.5 of Appendix M to this subpart, rounded off to the nearest 0.01.

(iii) The annual performance factors for water-source heat pumps shall be the annual performance factors determined according to section 5.6 of Appendix M to this subpart for the local ground water temperature, cooling load hours and heating load hours, rounded to the nearest 0.01.

(7) Other useful measures of energy consumption for central air conditioners shall be those measures of energy consumption which the Secretary of Energy determines are likely to assist consumers in making purchasing decisions and which are derived from the application of Appendix M to this subpart.

* * * * *

3. Section 430.23 is amended by establishing a new paragraph (m) to read as follows:

§ 430.23 Units to be tested

* * * * *

(m)(1) For central air conditioners, each condensing unit shall have a basic model selected and a sample of sufficient size tested in accordance with applicable provisions of this subpart such that:

(i) Any represented value of estimated annual operating cost, energy consumption or other measure of energy consumption of a basic model for which consumers would favor lower values shall be no less than the higher of (A) the mean of the sample or (B) the upper 90 percent confidence limit of the true mean divided by 1.05, and

(ii) Any represented value of the energy factor or other measure of energy consumption of a basic model for which consumers would favor higher values shall be no greater than the lower of (A) the mean of the sample or (B) the lower 90 percent confidence limit of the true mean divided by .95.

(2) The basic model [1] selected for tests shall be that condenser-evaporator-coil-combination likely to have the largest volume of retail sales. For every other basic model [1] using that condensing unit, either—

(i) A sample of sufficient size shall be tested to insure that:

(A) Any represented value of estimated annual operating cost, energy consumption or other measure of energy consumption of a basic model for which consumers would favor lower values shall be no less than the higher of (1) the mean of the sample or (2) the upper 90 percent confidence limit of the true mean divided by 1.05 and

(B) Any represented value of the energy factor or other measure of energy consumption of a basic model for which consumers would favor higher values shall be no greater than the lower of (1) the mean of the sample or (2) the lower 90 percent confidence limit of the true mean divided by .95, or

(ii) Represented values of measures of energy consumption shall be based on a computer simulation of the energy consuming characteristics of such basic model, as described in paragraph (m)(4) of this section.

(3) The sample selected for paragraph (m)(1) of this section shall be comprised of units which are production units, or representative of production units.

(4)(i) The basis of the computer simulation technique referred to in paragraph (2)(ii) of this section shall be a computer model of the mechanical vapor-Compression refrigeration cycle. The major components in the refrigeration cycle shall be modeled as "fits" to manufacturer component performance data. Heat transfer characteristics of the coils shall be modeled as functions of face area, number of rows, air flow rate, fins per inch, and entering air enthalpy. Compressor performance curves shall be used to provide refrigerant mass flow rate and compressor power as a function of suction temperature and condenser temperature. An interative mathematical technique shall be used which varies the evaporator temperature, condensing temperature, and refrigerant mass flow rate to balance the cycle.

(ii) The computer model shall be calibrated to actual test results derived pursuant to paragraph (m)(1) of this section and may be used to derive measures of energy consumption of other basic models which use the same condensing unit as the condenser-evaporator-coil-combination actually tested. Any manufacturer who chooses to rate products by computer simulation shall submit to DOE a document describing the analysis used to derive such ratings, including the computer model, and the data and engineering analysis used to verify the accuracy of such ratings.

(iii) Whenever measures of energy consumption determined by computer simulation do not agree with measures of energy consumption determined by actual testing, the values determined by actual testing shall be used to comply with section 323(c) of the Act, or to comply with rules prescribed under section 324 of the Act.

* * * * *

§ 430.24 [Revoked]

4. Section 430.24(m) is revoked.

5. Appendix M to Subpart B of Part 430 is amended to read as follows:

Appendix M To Subpart B—Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners

1. Definitions

1.1 "Annual performance factor" means the total heating and cooling done by a heat pump in a particular region in one year divided by the total electric power used in one year.

1.2 "ARI" means Air-Conditioning and Refrigeration Institute.

1.3. "ARI Standard 210–75" means the test standard published in 1975 by the ARI and titled "Standard for Unitary Air-Conditioning Equipment".

1.4 "ARI Standard 210–77" means the test standard published in 1977 by the ARI and titled "Standard for Air-Source Unitary heat Pump Equipment".

1.5 "ARI Standard 320–76" means the test standard published in 1976 by the ARI and titled "Standard for Water-Source Heat Pumps".

1.6 "ASHRAE" means the American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

1.7 "ASHRAE Standard 37–69" means the test standard published by ASHRAE in 1969 and titled "Methods of Testing for Rating Unitary Air-Conditioning and Heat Pump Equipment."

1.8 "Cooling load factor (CLF)" means the ratio of the total cooling done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady-state cooling done over the same period at constant ambient conditions.

1.9 "Cooling seasonal performance factor (CSPF)" means the estimated total cooling output of a unit during its normal annual usage period for cooling divided by the total electric power input during the same period.

1.10 "Cyclic Test" means a test where the indoor and outdoor conditions are held constant, but the unit is manually turned "on" and "off" for specific time periods to simulate part-load operation.

1.11 "Degradation coefficient (C_D)" means the measure of the efficiency loss due to the cycling of the unit.

1.12 "Demand-defrost control system" means a system which is designed to perform the defrost function on the outdoor coil of the heat pump only when a predetermined degradation of performance is sensed by measuring the difference between the outdoor air temperature and the saturated refrigerant temperature leaving the outdoor coil. The control system is able to defrost at all frosting temperature conditions while compensating for pressure differentials across the outdoor coil caused by factors other than defrosting.

1.13 "Design heating requirement (DHR)" is the amount of heating required to maintain a given indoor temperature at a particular outdoor design temperature.

1.14 "Dry-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will not condense on the evaporator coil of the unit.

1.15 "Heating seasonal performance factor (HSPF)" means the total hearing output of a heat pump during its normal annual usage period for heating divided by the total electric power input during the same period.

1.16 "Heating load factor (HLF)" means the ratio of the total heating done in a complete cycle of a specified time period, consisting of an "on" time and "off" time, to the steady state heating done over the same period at constant ambient conditions.

1.17 "Latent cooling" means the amount of cooling in Btu's necessary to remove water vapor from the air passing over the indoor coil by condensation during a period of time.

1.18 "Part-load factor (PLF)" means the ratio of the cyclic energy efficiency ratio to the steady-state energy efficiency ratio at identical ambient conditions.

1.19 "Seasonal energy efficiency ratio (SEER)" means the total cooling of a central

[1] Components of similar design may be substituted without requiring additional testing if the represented measures of energy consumption continue to satisfy the applicable sampling provision.

air conditioner in Btu's during its normal annual usage period for cooling divided by the total electric power input in watt-hours during the same period.

1.20 "Sensible cooling" means the amount of cooling in Btu's performed by a unit over a period of time, excluding latent cooling.

1.21 "Single package unit" means any central air conditioner in which all the major assemblies are enclosed in one system.

1.22 "Split system" means any central air conditioner in which one or more of the major assemblies are separate from the others.

1.23 "Steady-state test" means a test in which all indoor and outdoor conditions are held constant and the unit is in a continuous operating mode.

1.24 "Temperature bin" means a 5°F increment over a dry-bulb temperature range of 65°F through 104°F for the cooling cycle and −25°F through 64°F for the heating cycle.

1.25 "Time-temperature defrost control system" means a system which automatically provides the defrost function at a predetermined time interval whenever the outdoor temperature drops below a level where frosting will occur.

1.26 "Test condition tolerance" means the maximum permissible variation of the average of the test observations from the standard or desired test condition as provided in 6.1.1, 6.2.1, 6.2.2, and 6.2.3 of this Appendix.

1.27 "Test operating tolerance" means the maximum permissible difference between the maximum and the minimum instrument observation during a test as provided in 6.1.1, 6.2.1, 6.2.2 and 6.2.3 of this Appendix.

1.28 "Wet-coil test" means a test conducted at a wet-bulb temperature and a dry-bulb temperature such that moisture will condense on the test unit evaporator coil.

2. Testing Required

2.1 *Testing required for air source cooling only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* Two steady state wet coil tests required to be performed, test A and test B. Test A is to be conducted as an outdoor dry bulb temperature of 95°F and test B at 82°F. Tests C and D are optional tests to be conducted when cyclic performance parameters are to be measured in order to determine the degradation coefficient, C_D. Test C is a steady state dry coil test conducted at an outdoor dry bulb temperature of 82°F. Test D is a cyclic test also conducted at an outdoor dry bulb temperature of 82°F. In lieu of conducting tests C and D, an assigned value of 0.25 may be used for the degradation coefficient, C_D. Any manufacturer who chooses to use the assigned value must use it in calculations for all basic models.

2.1.1 *Testing required for units with single speed compressors and single speed condenser fans.* Test A and test B shall be performed according to the test procedures outlined in 4.1 of this Appendix. In addition, the cyclic performance shall be evaluated by conducting test C and D according to the requirements outlined in 4.1 of this Appendix.

2.1.2 *Testing required for units with single speed compressors and multiple-speed condenser fans.* The test requirements for multiple-speed condenser fan units shall be the same as described in section 2.1.1 for single speed condenser fan units.

2.1.3 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* The test requirements for two-speed compressor units, two compressor units, or units with cylinder unloading are the same as described in 2.1.1 of this Appendix except that test A and test B shall be performed at each compressor speed or at each compressor capacity.

2.1.4 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* When a unit employing a two-speed compressor, two compressors, or cylinder unloading provides a method of varying the ratio of the sensible cooling capacity to the total cooling capacity, (S/T), the test requirements are the same as for two-speed compressor units as described in 2.1.3 of this Appendix.

2.2 *Testing required for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cyclinder unloading.* Four types of tests are required to be performed: High Temperature, Cyclic, Frost Accumulation, and Low Temperature. In lieu of conducting the Cyclic Test an assigned value of 0.25 may be used for the degradation coefficient, C_D.

2.2.1 *Testing required for units with single speed compressors.* Units with single speed compressors shall be subjected respectively to the High Temperature Test at 47°F described in section 3.2.1.1, the Cyclic Test as described in section 3.2.1.2, the Frost Accumulation Test as described in section 3.2.1.3, and the Low Temperature Test as described in section 3.2.1.4. Any manufacturer who chooses to use the assigned value must use it in calculations for all basic models.

2.2.2 *Testing required for units with two-speed compressors, two compressors, or cylinder unloading.* With the unit operating: at the high compressors speed (two-speed compressor), with both compressors in operation (two compressors), or at the maximum capacity (cylinder unloading); the following tests are required to be performed on all units; the High Temperature Test at 47°F, the Frost Accumulation Test, and the Low Temperature Test. An additional test (cyclic at 47°F) is required, with the unit operating at the high compressor speed (two-speed compressor), with both compressors in operation (two compressors), or at the maximum capacity (cylinder unloading); if the normal mode of operation requires cycling "on" and "off" of the compressor(s) at high speed or maximum capacity.

With the unit operating: at the low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors), or at the low compressor capacity (cylinder unloading); the following tests are required to be performed on all units: the High Temperature Test at 47°F, the High Temperature Test at 62°F, and the Cyclic Test. Additional tests, (Frost Accumulation Test and Low Temperature Test) are required, with the unit operating: on low compressor speed (two-speed compressor), with the single compressor which normally operates at low loads (two compressors) or at the low compressor capacity (cylinder unloading), if the unit's low speed, one compressor or low capacity performance at and below 40°F is needed to calculate its seasonal performance.

2.3. *Testing required for air source units which provide both heating and cooling.* The requirements for units which provide both heating and cooling shall be the same as the requirements in Section 2.1 and 2.2 of this Appendix.

2.4 *Testing required for water source units which provide both heating and cooling.* The following four steady state tests, each employing two methods as described in 4.4, are required to be performed on all units: test E and test F as described in 3.4.1 and 3.4.3 of this Appendix and test G and test H as described in 3.4.2 and 3.4.3 of this Appendix.

2.4.1 *Cooling testing required for units with single speed compressors.* Two steady-state cooling tests, test E and test F, shall be performed according to the test procedures and test conditions outlined in 3.4 and 4.4 of this Appendix.

2.4.2 *Heating testing required for units with single speed compressors.* Two steady-state heating tests, test G and test H, shall be performed according to the test procedures and test conditions outlined in 3.4 and 4.4 of this Appendix.

2.4.3 *Cooling testing required for single speed compressor units capable of modulating water and refrigerant flow to maintain constant capacity.* The cooling tests required shall be the same as those detailed in 2.4.1 of this Appendix.

2.4.4 *Heating testing required for single speed compressor units capable of modulating water and refrigerant flow to maintain constant capacity.* The heating tests required shall be the same as those detailed in 2.4.2 of this Appendix.

3. Testing conditions

3.1 *Testing conditions for air source cooling only units with single speed compressors, two-speed compressors, two compressors or cylinder unloading.* The test room requirement and equipment installation procedures are the same as those specified in sections 11.1 and 11.2 of ASHRAE Standard 37–69. Units designed for either horizontal or vertical installation shall be tested in the vertical upflow position. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz), the test installation shall be designed such that there will be no air flow through the cooling coil due to natural or forced convection while the indoor fan is "off". This shall be accomplished by installing automatic dampers upstream and downstream of the test unit to block the off period air flow.

The following conditions listed in ARI Standard 210–75 shall apply to all tests performed in Section 3.1 of this Appendix:

5.1.1 Values of the Standard Capacity Ratings.

5.1.2 Values of the Standard Input Ratings.

5.1.3 Values of the Standard Energy Efficiency Ratio.
5.1.4.4 Cooling Coil Air Quantity.
5.1.4.5 Requirements for Separated Assemblies.

3.1.1 *Testing conditions for units with single speed compressors and single speed condenser fans.*

3.1.1.1 *Steady state wet-coil performance tests (Tests A and Test B).* Test A and test B shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80° F and a wet-bulb temperature of 67° F. The dry-bulb temperature of the air entering the outdoor side of the unit shall be 95° F in test A and 82° F in test B. The temperature of the air surrounding the outdoor side of the unit in each test shall be the same as the outdoor entering air temperature except for units or sections thereof intended to be installed only indoors, in which case the dry-bulb temperature surrounding that indoor side of the unit shall be 80° F. For those units which reject condensate to the condenser, located in the outdoor side of the unit, the outdoor wet-bulb temperature surrounding the outdoor side of the unit shall be 75° F in test A and 65° F in test B.

3.1.1.2 *Steady state dry coil performance test (Test C) and cyclic dry coil performance test (Test D).* Test C and test D shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80° F and a wet-bulb temperature which does not result in formation of condensate on the indoor coil. (It is recommended that an indoor wet-bulb temperature of 57° F or less be used.) The dry-bulb temperature of the air entering the outdoor portion of the unit shall be 82° F. The outdoor portion of the unit shall be subject to the same conditions as the requirements for conducting test B as stated previously in section 3.1.1.1. Test C shall be conducted with the unit operating steadily. Test D shall be conducted by manually cycling the unit "on" and "off." The unit shall cycle with the compressor "on" for 6 minutes and "off" for 24 minutes. The indoor fan shall also cycle "on" and "off", the duration of the indoor fan "on" and "off" periods being governed by the automatic controls which the manufacturer normally supplies with the unit. The results of tests C and D shall be used to calculate a degradation coefficient, C_D, by the procedures outlined in 5.1 of this Appendix.

3.1.2 *Testing conditions for units with single speed compressors and multiple-speed condenser fans.* The condenser fan speed to be used in test A shall be that speed which normally occurs at an outdoor dry-bulb temperature of 95° F, and for test B, the fan speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82° F. If elected to be performed, tests C and D shall be conducted at the same condenser fan speed as in test B.

3.1.3 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading.* The condenser fan speed used in conducting test A at each compressor speed shall be that which normally occurs at an outdoor dry-bulb temperature of 95° F. For test B, the condenser fan speed at each compressor speed shall be that which normally occurs at an outdoor dry-bulb temperature of 82° F. If elected to be performed, tests C and D shall be conducted at the low compressor speed with the same condenser fan speed as used in test B. For those two-speed units in which the normal mode of operation involves cycling the compressor "on" and "off" at high speed, tests C and D shall also be performed with the compressor operating at high speed and at a condenser fan speed that normally occurs at Test A ambient conditions. Units consisting of two compressors are subject to the same requirements as those units containing two-speed compressors, except that when operated at high speed, both compressors shall be operating and when operating at low speed, only the compressor which normally operates at an outdoor dry-bulb temperature of 82° F shall be operating.

In lieu of conducting tests C and D, an assigned value of 0.25 may b used for the degradation coefficient, C_D at each compressor speed. If the assigned degradation coefficient is used for one compressor speed it must also be used for the other compressor speed.

In the case of units with cylinder unloading, the loaded and the unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

3.1.4 *Testing conditions for units with two-speed compressors, two compressors, or cylinder unloading capable of varying the sensible to total (S/T) capacity ratio.* The mode of operation selected for controlling the S/T ratio in the performance of test A and test B at each compressor speed shall be such that it does not result in an operating configuration which is not typical of a normal residential installation. If elected to be performed, tests C and D shall be conducted at low compressor speed (single compressor operating) with the same S/T control mode as used in test B when performed at the low compressor speed. Likewise, tests C and D shall also be conducted at high compressor speed (two compressors operating) and with the same S/T control mode as in test A when performed at the high compressor speed.

In the case of units with cylinder unloading, the loaded and unloaded conditions correspond to high and low compressor speed on two-speed units respectively.

3.2 *Testing conditions for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* The equipment under test shall be installed according to the requirements of Section 11.2 of ASHRAE Standard 37–69 and Section 5.1.4.5 of ARI Standard 24077. Test chamber requirements are the same as given in Section 11.1 of ASHRAE Standard 37–39. Units designed for either horizontal or vertical installation shall be tested in the vertical upflow position. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 of 230 volts and 60 hertz).

3.2.1 *Testing conditions for units with single speed compressors.*

3.2.1.1 *High temperature test conditions.* The High Temperature Test at 47° F shall be conducted at an outdoor dry-bulb temperature of 47° F and an outdoor wet-bulb temperature at 43° F. The High Temperature Test at 62° F shall be conducted at an outdoor dry-bulb temperature of 62° F and an outdoor wet-bulb temperature of 56.5° F. For both tests, the dry-bulb air temperature entering and surrounding the indoor portion of the unit shall be 70° F and a maximum wet-bulb temperature of 60° F. The duration of the tests shall be for a minimum of ½ hour.

3.2.1.2 *Cycling test conditions.* The Cycling Test at 47° F shall be conducted at the same dry-bulb and wet-bulb temperature as the High Temperature Test at 47° F as described in 3.2.1.1. During the Cycling Test, the indoor fan shall cycle "on" and "off" as the compressor cycles "on" and "off", except that the indoor fan cycling times may be delayed due to controls that are normally installed with the unit. The compressor cycling times shall be 6 minutes "On" and 24 minutes "off." The test installation shall be designed such that there will be no airflow through the indoor unit due to natural or forced convection while the indoor fan is "off." This shall be accomplished by installing automatic dampers upstream and downstream of the test unit to block the off period airflow.

3.2.1.3 *Frost accumulation test conditions.* The dry-bulb temperature and dew-point temperature of the air entering the outdoor portion of the unit shall be 35° F and 30° F respectively. The indoor dry-bulb temperature shall be 70° F and the maximum indoor wet-bulb temperature shall be 60° F. The Frost Accumulation Test requires that the unit undergo a defrost prior to the actual test. The test then begins at defrost termination and ends at the next defrost termination. Defrost termination occurs when the controls normally installed within the unit are actuated to cause it to change defrost operation to normal heating operation. During the test, auxiliary resistance heaters shall not be employed during either the heating or defrost portion of the test.

3.2.1.4 *Low temperature test conditions.* The Low Temperature Test shall be conducted at a dry-bulb temperature entering the outdoor portion of the unit of 17° F and a wet-bulb temperature of 15° F. The air entering the indoor portion of the unit shall have a dry-bulb temperature of 70° F and a maximum wet-bulb temperature of 60° F.

3.2.1.5 *Additional testing conditions.* All tests shall be conducted at the indoor-side air quantities specified in Sections 5.1.4.3 and 5.1.4.6 and Table 2 of ARI Standard 240–77. The following conditions listed in ARI Standard 240–77 shall apply to all tests performed in Section 3.2 of this Appendix:

5.4.4.4 *Outdoor-Side Air Quantity; and*
5.1.4.5 *Requirements for Separated Assemblies.*

In all tests, the specified dry-bulb temperature entering the outdoor portion of the unit also applies to the air temperature surrounding the outdoor portion of the unit. Similarly, models where portions are

intended to be installed indoors shall have the air temperature surrounding that poriton of the unit the same as the indoor air temperature.

3.2.2 *Testing conditions for units with two speed compressors, two compressors or cylinder unloading.* The testing conditions for two speed compressors, two compressors, or cylinder unloading shall be the same as those for single speed units as described in 3.2.1.

3.3 *Testing conditions for air source units which provide both heating and cooling.* The testing conditions for units which provide both heating and cooling shall be the same as the requirements in Sections 3.1 and 3.2 of this Appendix.

3.4 *Testing conditions for water source units which provide both heating and cooling.* The test room requirements and equipment installation procedures are the same as those specified in Sections 11.1 and 11.2 of ASHRAE Standard 37–69. All tests shall be performed at the normal residential voltage and frequency for which the equipment is designed (either 115 or 230 volts and 60 hertz).

3.4.1 *Cooling testing conditions for units with single speed compressors.* Test E and test F shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 80° F and a wet-bulb temperature of 67° F. The water temperature entering the refrigerant-to-water heat exchanger on the outdoor side of the unit shall be 85° F for test E and 60° F for test F. The water temperature leaving the refrigerant-to-water heat exchanger on the outdoor side of the unit shall be 95° F in test E. For test F, the temperature shall be one which corresponds to the same flow rate as determined in test E. The temperature of the air surrounding the unit in each test shall be 80° F.

3.4.2 *Heating testing conditions for units with single speed compressors.* Test G and test H shall be performed with the air entering the indoor side of the unit having a dry bulb temperature of 70° F and a maximum wet bulb temperature of 60° F. The water temperature entering the refrigerant-to-water heat exchanger on the outdoor side of the unit shall be 80° F for test G and 60° F for test H. The water flow rate shall be the same as determined from test E and F in 3.1.1 of this Appendix. The temperature surrounding the unit in each test shall be 70° F.

3.4.3 *Cooling testing conditions for single speed compressor units capable of modulating water and refrigerant flow to maintain constant capacity.* The same conditions as detailed in 3.4.1 shall apply except that no constraint shall be placed on the outlet water temperature leaving the refrigerant-to-water heat exchanger in test E and test F.

3.4.4 *Heating testing conditions for single speed compressor units capable of modulating water and refrigerant flow to maintain constant capacity.* Test G and test H shall be performed with the air entering the indoor side of the unit having a dry-bulb temperature of 70° F and a maximum wet-bulb temperature of 60° F. The water temperature entering the refrigerant-to-water heat exchanger on the outdoor side of the unit shall be 80° F with test G and 60° F for test H. The water flow rate shall be determined by the controls set by the manufacturer. The control shall not be adjusted between the tests on the cooling performance and the test on the heating performance.

4.0 *Testing procedures.* Measure all electrical inputs as described in the procedures below. All electrical measurements shall include auxiliary power or energy (controls, transformers, crankcase heaters, etc.) delivered to the unit.

4.1 *Testing procedures for air source cooling only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.* All steady-state wet- and dry-coil performance tests on single package units shall simultaneously employ the Air-Enthalpy Method (Section 3 of ASHRAE Standard 37–69) on the indoor side and one other method consisting of either the Air-Enthalpy Method or the Compressor Calibration Method (Section 4 of ASHRAE Standard 37–69) on the outdoor side. All steady-state wet- and dry-coil performance tests on split systems shall simultaneously employ the Air-Enthalpy Method or the Compressor Calibration Method on the indoor side and the Air-Enthalpy Method, the Compressor Calibration Method, or the Volatile Refrigerant Flow Method (Section 5 of ASHRAE Standard 37–69) on the outside. All cyclic dry-coil performance tests shall employ the Air-Enthalpy Method, indoor side only. The values calculated from the two test methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calculations in Section 5.1. Units shall be installed and tested in such a manner that when operated under steady-state conditions, the cooling coil and condenser coil air flows meet the requirements of Sections 5.1.4.3, 5.1.4.4, and 5.1.4.6 of Standard 210–75.

4.1.1 *Test operating procedures.*

4.1.1.1 *Steady-state wet-coil performance tests (Test A and Test B).* Steady-state wet-coil performance tests (A and B) shall be conducted in accordance with the conditions described in sections 3.1.1.1, 3.1.2, 3.1.3, 3.1.4, and 3.1.5 of this Appendix and the procedures described for cooling tests in Section 11.3 of ASHRAE Standard 37–69 and evaluated in accordance with the cooling-related requirements of Section 12 of the ASHRAE Standard 37–69. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained.

4.1.1.2 *Steady-state and cyclic dry-coil performance tests (Test C and Test D).* The steady-state and cyclic dry-coil tests (C and D) shall be conducted as described below in accordance with the conditions described in sections 3.1.1.2, 3.1.2, 3.1.3, 3.1.4, and 3.1.5 of this Appendix. The results shall be evaluated in accordance with the cooling related requirements of Sections 12.1.5, 12.1.6, 12.1.7, of ASHRAE Standard 37–69. The test room reconditioning apparatus and the equipment under test shall be operated until equilibrium conditions are attained, but not for less than one hour before data for test C are recorded. For all equipment test methods including the Compressor Calibration Method, test C shall be performed with data recorded at 10-minute intervals until four consecutive sets of readings are attained with the tolerance prescribed in Section 11.6 of ASHRAE Standard 37–69. When the Air-Enthalpy Method is used on the outdoor side for test C, the requirements of this section shall apply to both the preliminary test and the regular equipment test; the requirements of Section 3.6 of ASHRAE Standard 37–69 shall also apply. Immediately after test C is completed, the test unit shall be manually cycled "off" and "on" using the time periods from 3.1.1 of this Appendix until steadily repeating ambient conditions are again achieved in both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be run through an additional "off"/"on" cycle during which the test data required in 5.1 shall be recorded. During this last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerances specified in 4.1.3 of this Appendix during the cyclic dry-coil tests, all air moving equipment on the condenser side shall cycle "on" and "off" when the compressor cycles "on" and "off". The indoor air moving equipment shall also cycle "off" as governed by any automatic controls normally installed with the unit. This last requirement applies to units having an indoor fan time delay. Units not supplied with an indoor fan time delay shall have the indoor air moving equipment cycle "on" and "off" as the compressor cycles "on" and "off."

4.1.2 *Test instrumentation.* The steady-state and cyclic performance tests shall have the same requirements pertaining to instrumentation and data as those specified in Section 10 and Table II of ASHRAE Standard 37–69. For the cyclic dry-coil performance tests, the dry-bulb temperature of the air entering and leaving the cooling coil, or the difference between these two dry-bulb temperatures, shall be continuously recorded with instrumentation accurate to within ± 0.3° F of indicated value and have a response time of 2.5 seconds or less. Response time in the time required for the instrumentation to obtain 63 percent of the final steady-state temperature difference when subjected to a step change in temperature difference of 15° F or more. Electrical measurement devices (watt-hour meters) used during all tests shall be accurate to within ± 0.5 percent of indicated value.

4.1.3 *Test tolerances.* All steady-state wet- and dry-coil performance tests shall be performed within the applicable operating and test condition tolerances specified in Section 11.6 and Table III of ASHRAE Standard 37–69.

4.1.3.1 The indoor and outdoor average dry-bulb temperature for the cyclic dry coil test D shall both be within 1.0° F of the indoor and outdoor average dry bulb temperature for the steady-state dry coil test C.

4.1.3.2 The test condition and test operating tolerances for conducting test D are stated in 6.1.1 of this Appendix. Variation in

the test conditions greater than the tolerances prescribed in 6.1.1 of this Appendix shall invalidate the test. It is suggested that an electric resistance heater having a heating capacity approximately equal to the sum of the cooling capacity and compressor and condenser fan power should be installed in the outdoor test room and cycled "off" and "on" as the unit cycles "on" and "off" respectively to improve control in the outdoor test room. Similarly, an electric resistance heater having a heating capacity approximately equal to the cooling capacity of the unit could be installed in the indoor test room, and cycled "on" and "off" as the test unit cycles "on" and "off" to improve indoor room control.

4.2 *Testing procedures for air source heating only units with single speed compressors, two-speed compressors, two compressors, or cylinder unloading.*

4.2.1. *Test operating procedures.* All High Temperature Tests, the Cyclic Test, the Frost Accumulation Test, and the Low Temperature test shall have the performance evaluated by the Air-Enthalpy Method on the indoor side. In addition, the High Temperature Test and the Low Temperature Test shall have a simultaneous test method (as described in 4.1) used as a check. The values calculated from the two methods must agree within 6 percent in order to constitute a valid test. Only the results from the Air-Enthalpy Method on the indoor side shall be used in the calculations in section 5.2.

4.2.1.1 *Test procedure for high temperature test.* When the outdoor Air-Enthalpy Method isused, the outdoor chamber must not interfere with the normal air circulating pattern during the preliminary test. It is necessary to determine and adjust for system resistance when the outdoor air measuring apparatus is attached to the outdoor portion of the unit. The test room apparatus and test units must be operated for at least one hour with at least ½ hour at equilibrium and at the specified test conditions prior to starting the test. The High Temperature Test shall then be conducted for a munimum of ½ hour with intermittent data being recorded at 10-minute intervals. For all units, especially those having controls which periodically cause the unit to operate in defrost mode, attention should be given to prevent defrost during the High Temperature Test. Units which have undergone a defrost should operate in the heating mode for at least 10-minutes after defrost temination prior to the start of the test. When the outdoor Air-Enthalpy Method is used as a second test, then a preliminary test must be conducted for a minimum of 30 minutes with 4 or more sets of data recorded at 10 minute intervals, all remaining requirements of Section 3.6.1 in the ASHRAE Standard 37–69 shall then apply in conducting the preliminary test for the outdoor air enthalpy method. For some units, at the ambient condition of the test, frost may accumulate on the outdoor coil. If the supply air temperature or the difference between the supply air temperature and the indoor air entering temperature has decreased by more than 1.5° F at the end of the test, the unit shall be defrosted and the test restarted. Only the results of this second High Temperature Test shall be used in the heating seasonal performance calculation in section 5.2. Prior to beginning the High Temperature Test, a unit shall operate in the heating mode for at least 10 minutes after defrost termination to establish equilibrium conditions for the unit and the room reconditioning apparatus. The High Temperature Test may only begin when the test unit and room conditions are within the test condition tolerances specified in Section 6.2.1 of this Appendix.

4.2.1.2 *Test procedures for the cyclic test.* The cyclic test shall follow the High Temperature Test and be cycled "on" and "off" as specified in 3.2.1.2 until steadily repeating ambient conditions are achieved for both the indoor and outdoor test chambers, but for not less than 2 complete "off"/"on" cycles. Without a break in the cycling pattern, the unit shall be operated through an additional "off"/"on" cycle, during which the required test data shall be recorded. During the last cycle, which is referred to as the test cycle, the indoor and outdoor test room ambient conditions shall remain within the tolerance specified in section 6.2.2. of this Apendix. If, prior to the High Temperature Test, the unit underwent a defrost cycle to rid the outdoor coil of any accumulated frost, then prior to cycling the unit "off" and "on" it should be made to undergo a defrost. After defrost is compeleted and before starting the cycling process, the unit shall be operated continuously in the heating mode for at least 10 minutes to assure that equilibrium conditions have again been established for the unit and the room conditioning apparatus. Cycling the unit may begin when the test unit and room conditions are within the High Temperature Test condltion tolerances specified in section 6.2.1 of this Appendix. Attention should be given to prevent defrost after the cycling process has begun.

4.2.1.3 *Test procedures for the frost accumulation test.* The defrost controls shall be set at the normal settings which most typify those encountered in Region IV as described in section 6.2.4 and 6.2.5 of this Appendix. The test room reconditioning equipment and the unit under test shall be operated for at least ½ hour prior to the start of a "preliminary" test period. The preliminary test period and the test period itself are to be conducted within the test tolerances given in section 4.2.3.3 of this Appendix. In some cases, the preliminary defrost cycle may be manually induced, however, it is important that the normally operating controls govern the defrost termination in all cases. For units containing defrost controls which are likely to cause defrost at intervals less than one hour when the unit is operating at the required test conditions, the preliminary test period shall start at the termination of a defrost cycle which automatically occurs and shall end at the termination of the next automatically occuring defrost cycle. For units containing defrost controls which are likely to cause defrost at intervals exceeding one hour when operating at the required test condition, the preliminary test period consists of "heating-only" preliminary operation for at least one hour, after which a defrost may be manually or automatically induced. The test period then begins at the termination of this defrost cycle and ends at the termination of the next automatically occurring defrost cycle. If the unit has not undergone a defrost after 12 hours, then the tests shall be concluded and the results calculated for this 12-hour period. For units which turn the indoor fan off during defrost, the indoor supply duct shall be blocked during all defrost cycles to prevent natural or forced convection through the indoor unit. During defrost, resistance heaters normally installed with the unit shall be prevented from operating.

4.2.1.4 *Test procedures for the low temperature test.* Where applicable, the High Temperature Test preparation and performance requirements shall also be used in the Low Temperature Test. The test room reconditioning equipment shall first be operated in a steady-state manner for at least one-half hour at equilibrium and at the specified test conditions. The unit shall then undergo a defrost, either automatic or manually induced. It is important that the unit terminate the defrost sequence by the action of its own defrost controls. The defrost controls are to remain at the same setting as specified in 4.2.1.3. At a time no earlier than 10 minutes after defrost termination, the test shall start. Test duration is one-half hour. For all units, defrost should be prevented during the one-half hour test period.

4.2.2 *Test instrumentation.*

4.2.2.1 *Test instrumentation for the high temperature test:* The indoor air flow rate shall be determined as described in Section 7.1 through 7.4 of ASHRAE Standard 37–69. This requires the construction of an air receiving chamber and discharge chamber separated by partition in which one or more nozzles are located. The receiving chamber is connected to the indoor air discharge side of the test specimen through a short plenum. The exhaust side of the air flow rate measuring device contains an exhaust fan with some means to vary its capacity to obtain the desired external resistance to air flow rate. The exhaust side is then left open to the test room or is ducted through a conditioning apparatus and then back to the test specimen inlet. The static pressure across the nozzles, the veolcity pressure, and the static pressure measurements at the nozzle throat shall be measured with manometers which will result in errors which are no greater than ±1.0 percent of indicated value and having minimum scale divisions not exceeding 2.0 percent of the reading. Static pressure and temperature measurements must be taken at the nozzle throat in order to obtain density of the air. The areas of the nozzles shall be determined by measuring their diameter with an error no greater than ±0.2 percent in four places approximately 45 degrees apart, around the nozzle in each of two places through the nozzle throat, one at the outlets and the others in the straight section near the radius. The energy usage of the compressor, indoor and outdoor fan, and all other equipment components shall be measured with a watt-hour meter which is accurate to within ±0.5 percent of the quantity measured. Measurements of the air temperature entering

and leaving the indoor coil or the difference between these two shall be made in accordance with the requirements of ASHRAE Standard 41 part 1. These temperatures shall be continuously recorded with instrumentation having a total system accuracy within ±0.3°F of indicated value and a response time of 2.5 seconds or less. Temperature measurements are to be made upstream of the static pressure tap on the inlet and downstream of the static pressure taps on the outlet. The indoor and outdoor dry-bulb temperatures shall be continuously recorded with instrumentation which will result in an error no greater than ±0.3°F of indicated value. The outdoor wet-bulb temperature shall be continuously recorded. Static pressure measurements in the ducts and across the unit shall be continuously recorded. Static pressure measurements in the ducts and across the unit shall be made in accordance with Section 8 of ASHRAE Standard 37–69 using a manometer which will result in an error no greater than ±0.01 inch of water. All other data not continuously recorded shall be recorded at 10 minute intervals.

4.2.2.2 *Test instrumentation for the cycling test.* The air flow rate during the on-period of the Cyclic Test shall be the same agree within ±1. percent) as the air flow rate measured during the previously conducted High Temperature Test. All other instrumentation requirements are identical to 4.2.2.1 of this Appendix.

4.2.2.3 *Test instrumentation for the frost accumulation test.* The air flow rate for the Frost Accumulation Test shall be the same as described in 4.2.2.1. The indoor-side dry-bulb temperature and outdoor-side dry-bulb temperature shall be continuously recorded with instrumentation having a total system accuracy within ±0.3°F of indicated value. The outdoor dew point temperature shall be measured with an error no greater than ±0.5°F of indicated value using continuously recording instrumentation. All other data shall be continuously recorded at 10 minute intervals during the heating cycle. Defrost initiation, termination and complete test cycle time (from defrost termination to defrost termination) shall be recorded. Defrost initiation is defined as the actuation (either automatically or manually) of the controls normally installed with the unit which cause it to alter its normal heating operation in order to eliminate possible accumulations of frost on the outdoor coil. Defrost termination occurs when the controls normally within the unit are actuated to change from defrost operation to normal heating operation. Provisions should be made so that instrumentation is capable of recording the cooling done during defrost as well as the total electrical energy usage during defrost. These data and the continuously recorded data need be the only data obtained during defrost.

4.2.2.4 *Test instrumentation for the low temperature test.* Instrumentation for the Low Temperature Test is identical to that of the High Temperature Test described in section 4.2.2.1 of this Appendix.

4.2.3 *Test tolerances.*

4.2.3.1 *Test tolerances for the high temperature test.* All tests shall be conducted within the tolerances specified in Section 6.2.1. Variation greater than those given shall invalidate the test. The heating capacity results by the indoor Air Enthalpy Method shall agree within 6 percent of the value determined by any other simultaneously conducted capacity test in order for the test to be valid.

4.2.3.2 *Test tolerances for the cyclic test.* The test condition tolerances and test operating tolerances for the on-period portion of the test cycle are specified in Section 6.2.2. Variation exceeding any specified test tolerance shall invalidate the test results.

4.2.3.3 *Test tolerances for the frost accumulation test.* Test condition and test operating tolerances for Frost Accumulation Tests are specified in Section 6.2.3. Test operating tolerances during heating applies when the unit is in the heating mode, except for the first 5 minutes after the termination of a defrost cycle. Test operating tolerance during defrost applies during a defrost cycle and during the first 5 minutes after defrost termination when the unit is operating in the heating mode. In determining whether the test condition tolerances are met, only the heating portion of the test period shall be used in calculating the average values. Variations exceeding the tolerances presented in Section 6.2.3 shall invalidate the test.

4.2.3.4 *Test tolerances for the low temperature test.* During the test period for the Low Temperature Test, the operating conditions shall be within the tolerances specified in Section 6.2.1 of this Appendix.

4.3 *Testing procedures for air source units which provide both heating and cooling.* The testing procedures for units which provide both heating and cooling shall be the same as those specified in Section 4.1 and 4.2 of this Appendix. Also during the off-period of the dry-coil cooling test (test D), the switch-over valve shall remain in the cooling mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve. During the off period of the cyclic heating test at 47°F, the switch-over valve shall remain in the heating mode, unless the controls normally supplied with the unit are designed to reverse it, in which case the controls shall operate the valve.

4.4 *Testing procedures for water source units which provide both heating and cooling.* All steady-state heating and cooling performance tests on units shall simultaneously employ the Air Enthalpy Method (Section 6 of ASHRAE Standard 37–69) on the outdoor side.

4.4.1 *Test operating procedures.* All heating and cooling performance tests shall be conducted in accordance with the procedures in Section 11 of ASHRAE Standard 37–69 and evaluated in accordance with the related requirements of Section 6 and 12 of ASHRAE Standard 37–69.

4.4.2 *Test instrumentation.* The heating and cooling performance tests shall have the same requirements pertaining to instrumentation and data as those specified in Section 10 and Table II of ASHRAE Standard 37–69. Units shall be installed and tested in such a manner that when operated under steady-state conditions, the cooling coil and condenser air flows meet the requirements of Sections 5.1.4.3 and 5.1.4.5 of ARI Standard 320–76.

4.4.3 *Test tolerances.* All heating and cooling performance tests shall be performed within the applicable operating and test condition tolerances specified in Section 11.6 and Table III of ASHRAE Standard 37–69.

5.0 *Calculations for performance factors.*

5.1 *Calculations of cooling seasonal performance factors (CSPF) and seasonal energy efficiency ratios (SEER) in air-source units.*

The testing data and results required for these calculations shall include the following:

(i) Cooling capacities (Btu/hr) from tests A and B and, if applicable, the cooling capacity (Btu/hr) from test C and the total cooling done from test D (Btu's).

$\dot{Q}_{ss}^{k}$ (95F)

$\dot{Q}_{ss}^{k}$ (82F)

$\dot{Q}_{ss}$, dry

Q_{cyc}, dry

(ii) Electrical power input to all components and controls (watts) from tests A, B, and if applicable the electrical power input to all components and controls (watts) from test C and the electrical usage (watt-hour) from test D.

$\dot{E}_{ss}^{k}$ (95F)

$\dot{E}_{ss}^{k}$ (82F)

$\dot{E}_{ss}^{k}$, dry

E_{cyc}, dry

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water).

(iv) Air temperature (°F)

Outdoor dry bulb

Outdoor wet bulb

Indoor dry bulb

Indoor wet bulb

Where the cooling capacities $\dot{Q}_{ss}^{k}$ (95F), from test A, $\dot{Q}_{ss}^{k}$ (82), from test B, and $\dot{Q}_{ss}$, dry, from test C, are calculated using the equations specified in section 3.7 of ASHRAE Standard 37–69. The total cooling done, Q_{cyc}, dry from test D, is calculated using equation (1) below.

Units which do not have indoor air circulating fans furnished as part of the model shall have their measured total cooling capacities adjusted by subtracting 1250 Btu/hr per 1,000 CFM of measured indoor air flow and adding to the total steady-state electrical power input 365 watts per 1,000 CFM of measured indoor air flow.

Energy efficiency ratios from tests A, B, and C, EER_A, EER_B, and EER_{ss}, dry respectively, are each calculated as the ratio of the total cooling capacity in Btu/hr to the total electrical power input in watts.

Units which do not have indoor air circulating fans furnished as part of the model shall adjust their total cooling done and energy used in one complete cycle for the effect of circulating indoor air equipment power. The value to be used for the circulating indoor air equipment power shall be 1250 Btu/hr per 1,000 CFM of circulating indoor air. The energy usage required in one complete cycle required for indoor air circulation is the product of the circulating

indoor air equipment power and the duration of time in one cycle that the circulating indoor air equipment is on. The total cooling done shall then be the measured cooling in one complete cycle minus the energy usage required for indoor air circulation in one complete cycle. The total electrical energy usage shall be the sum of the energy usage required for indoor air circulation in one complete cycle and the energy used by the remaining equipment components (compressor(s), outdoor fan, crankcase heater, transformer(s), etc.) in one complete test cycle.

Energy efficiency ratio from tests D, EER_{cyc} dry is calculated as the ratio of the total cooling done in Btu's to the total electrical energy usage in watt-hours.

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The results of the cyclic and steady-state dry-coil performance tests shall be used in the following (4) equations:

$$(1) \qquad Q_{cyc,\,dry} = \frac{60 \times \dot{V} \times C_{pa} \times}{[V_n' \times (1 + W_n)]}$$

where

$Q_{cyc,dry}$ = Total cooling over a cycle consisting of one compressor "off" period and one compressor "on" period (Btu's)

$\dot{V}$ = Indoor air flow rate (cfm) at the dry-bulb temperature, humidity ratio, and pressure existing in the region of measurement.

C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air. (Btu/lb-°F).

V_n' = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft^3/lb).

W_n = Humidity ratio (lb/lb).

and Γ (hr.-°F), which is described by the equation:

(2)

$$\Gamma = \int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}} [T_{a1}(t) - T_{a2}(t)]\,dt$$

where

$T_{a1}(t)$ = Dry bulb temperature of air entering the indoor coil (°F) at time (t).

$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

$$(3) \qquad CLF = \frac{Q_{cyc,\,dry}}{Q_{ss\ dry} \times \tau}$$

where

CLF = cooling load factor.

$Q_{ss\ dry}$ = Total steady-state cooling capacity from test C (Btu/hr).

τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

The proceding equations are then used in the following equation to calculate a degradation coefficient C_D rounded to the nearest .01.

$$C_D = \frac{1 - \frac{EER_{cyc,dry}}{EER_{ss,dry}}}{1 - CLF}$$

(4)

where

$EER_{ss,dry}$ = Energy efficiency ratio from test C, (Btu/watt-hr).

Calculate the cooling seasonal performance factor (CSPF) for air source central air conditioner or heat pump.

$$CSPF = \frac{SEER}{3.413}$$

where:

SEER = the seasonal energy efficiency ratio as determined in section 5.1.1, 5.1.2, 5.1.3, or 5.1.4 below.

3.413 = a conversion factor which converts watt-hours to Btu's.

5.1.1 Method for calculating a SEER for units with single-speed compressor and single speed condenser fans. The seasonal energy efficiency ratio for units employing single-speed compressors and single-speed condenser fans shall be based on the performance of test B and a method outlined in 2.1.1 and 3.1.1 of this Appendix to account for the cyclic performance.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

$$SEER = PLF\ (0.5) \times EER_B$$

where

EER_B = Energy efficiency ratio determined from test B as outlined in 2.1.1

PLF(0.5)= Part-load performance factor determined from the equation:

$$PLF\ (0.5) = 1 - C_D\ (1-0.5)$$

where

C_D = Is the degradation coefficient described in 2.1.1 or as calculated in equation (4) above

5.1.2 Method for calculating a SEER for units with single-speed compressors and multi-speed condenser fans. The seasonal energy efficiency ratio (SEER) for units employing single-speed compressors and multi-speed condenser fans shall be based on the energy efficiency ratio obtained for test B and the method outlined in 2.1.2 of this Appendix to account for the performance under cyclic conditions. The energy efficiency ratio for test B is obtained with the unit operating with the condenser fan speed which normally occurs at test B ambient conditions.

The seasonal energy efficiency ratio in Btu's/watt-hour shall be determined by the equation:

$$SEER = PLF\ (0.5) \times EER_B$$

where

EER_B = energy efficiency ratio determined from test B in 2.8

PLF(0.5) = Part-load performance factor as determined from the equation:

$$PLF\ (0.5) = 1 - C_D\ (1 - 0.5)$$

where

C_D = The degradation coefficient described in 2.1.2 or as calculated in equation (4) above.

5.1.3 Method for calculating a SEER for units with two speed compressors or two compressors, or cylinder unloading. The calculation procedure described in this section shall be based on the performance of test A and B at each of the compressor speeds for two-speed compressor units, subject to the conditions on condenser fan speed described in 3.1.3.

Units operating with two compressors shall have the SEER calculated in the same manner as two-speed compressor units. The superscripted index k=1, (and the term "low-speed") designates the compressor that normally operates at an outdoor dry-bulb temperature of 82°F and k=2 (and the term "high speed") denotes operation with both compressors.

In order to evaluate the steady-state capacity $Q_{ss}^k(T_j)$, and power input, $E_{ss}^k(T_j)$, at temperature T_j for each compressor speed, k=1, k=2, the results of tests A and B from 5.1 shall be used in the following equation:

$$Q_{ss}^k(T_j) = Q_{ss}^k(95\ F) + \frac{Q_{ss}^k(82\ F) - Q_{ss}^k(95\ F)}{95 - 82}\ [33-(5 \times j)]$$

where

$Q_{ss}^k(95\ F)$ = Steady-state capacity measured from test A as outlined in 2.1.3.

$Q_{ss}^k(82\ F)$ = Steady-state capacity measured from test B as outlined in 2.1.3.

$$E_{ss}^k(T_j) = E_{ss}^k(95\ F) + \frac{E_{ss}^k(82\ F) - E_{ss}^k(95\ F)}{95 - 82}\ [33 - (5 \times i)]$$

when

$E_{ss}^k(95\ F)$ = Electrical power input measured using test A as outlined in 2.1.3.

$E_{ss}^k(82\ F)$ = Electrical power input measured from using test A as outlined in 2.1.3.

The building cooling load, $BL(T_j)$ for the four cases described in section 5.1.3.1 through 5.1.3.4 shall be obtained from the following equation:

$$BL(T_j) = \frac{(5Xj) - 3}{95 - 65} \times \frac{Q_{ss}^{k=2}(95\ F)}{1.1}$$

where

$Q_{ss}^{k=2}(95\ F)$ = Steady-state capacity measured from test A in 2.9 at the high compressor speed.

The value of the degradation coefficient $C_D^{k=1}$ for low compressor speed cycling and $C_D^{k=2}$ for high speed on/off compressor cycling is determined as described in section 2.1.3, or as calculated above in equation (1).

5.1.3.1 Units operating at low compressor speed (k=1) for which the steady-state cooling capacity, $Q_{ss}^{k=1}(T_j)$, is greater than or equal to the building cooling load, $BL(T_j)$, evaluate the following equations:

(1) $$X^{k=1} = \frac{BL(T_j)}{Q_{ss}^{k=1}(T_j)}.$$

where

$X^{k=1}$ = Load factor.

$BL(T_j)$ = Building cooling load (Btu/hr) at temperature (T_j) from section 5.1.3

$Q_{ss}^{k=1}(T_j)$ = Steady-state cooling capacity (Btu/hr) at temperatures (T_j) from section 5.1.3.

(2) $\frac{Q(T_j)}{N} = X^{k=1} \times Q_{ss}^{k=1}(T_j) \times \frac{n_j}{N}$

where

$\frac{Q(T_j)}{N}$ = ratio of total cooling (Btu) in temperature bin j to the number of temperature bin hours.

$\frac{n_j}{N}$ is the fractional number of hours in temperature bin j from 6.1.2

(3) $\frac{E(T_j)}{N} = \frac{X^{k=1} \times E_{ss}^{k-1}(T_j)}{PLF^{k=1}} \times \frac{n_j}{N}$

where

$\frac{E(T_j)}{N}$ = ratio of Energy usage (watt-hr) in temperature bin j to the number of temperature bin hours.

$PLF^{k=1} = 1 - C_D^{k=1}(1 - X^{k=1})$.

Where C_D the degradation coefficient as described in section 2.1.3 or as calculated above in equation (1)

5.1.3.2 When a unit must alternate between high (k=2) and low (k=1) compressor speeds to satisfy the building cooling load at a temperature Tj, evaluate the following equations:

(1) $X^{k=1} = \frac{Q_{ss}^{k=2}(T_j) - BL(T_j)}{Q_{ss}^{k=2}(T_j) - Q_{ss}^{k=1}(T_j)}$

(2) $X^{k=2} = 1 - X^{k=1}$

(3) $\frac{Q(T_j)}{N} = [X^{k=1} \times Q_{ss}^{k=1}(T_j) + X^{k=2} \times Q_{ss}^{k=2}(T_j)] \times \frac{n_j}{N}$

(4) $\frac{E(T_j)}{N} = [X^{k=1} \times E_{ss}^{k=2}(T_j) + X^{k=2} \times E_{ss}^{k=2}(T_j)] \times \frac{n_j}{N}$

5.1.3.3 When a unit must cycle on and off at high compressor speed (k=2) in order to satisfy the building cooling load at a temperature Tj, evaluate the equations provided in section 5.1.3.1 replacing (k=1) data with the (k=2) data.

5.1.3.4 When a unit operates continuously at high compressor speed (k=2) at an outdoor temperature Tj evaluate the following equations:

(1) $\frac{Q(T_j)}{N} = Q_{ss}^{k=2}(T_j) \times \frac{n_j}{N}$

(2) $\frac{E(T_j)}{N} = E_{ss}^{k=2}(T_j) \times \frac{n_j}{N}$

5.1.3.5 Calculate the SEER in Btu's/watt-hr, using the values for the terms

$$\frac{Q(T_j)}{N}$$

and

$$\frac{E(T_j)}{N}$$

as determined at each temperature bin according to the applicable conditions described in sections 5.1.3.1 through 5.1.3.4 as follows:

$$SEER = \frac{\sum_{j=1}^{8} \frac{Q(T_j)}{N}}{\sum_{j=1}^{8} \frac{E(T_j)}{N}}$$

5.1.4 Method for calculating a SEER for units with two speed compressor, two compressor or cylinder unloading capable of varying the sensible to total capacity (S/T) ratio. Multi-speed compressor and two-speed compressor units capable of varying the sensible to total capacity ratio (S/T) shall have the seasonal energy efficiency ratio determined as described in section 5.1.3. For such units, the mode of operation selected to determine the steady-state capacities $Q_{ss}^{k}(95)$, $Q_{ss}^{k}(82)$, $E_{ss}^{k}(95)$, $E_{ss}^{k}(82)$, and power inputs at each compressor speed k=1, k=2, for tests A and B is outlined in section 2.10

5.2 Calculation of Heating Seasonal Performance Factors (HSPF) for Air-Source Units.

The testing data and results required for these calculations shall include the following:

(i) Heating capacities (Btu/hr) from the indoor air enthalpy method for the High Temperature Tests, and the total heating done (Btu's) for the cyclic and frost accumulation tests

$\dot{Q}_{ss}$ (47) or $\dot{Q}_{ss}$ (62)

$\dot{Q}_{ss}$ (17)

Q_{cyc} (47)

Q_{DEF} (35)

(ii) Electrical power input to all components (watts) for the steady state tests, and the electrical usage (watt-hours) for the cyclic and frost accumulation tests

$\dot{E}_{ss}$ (47) or $\dot{E}_{ss}$ (62)

$\dot{E}_{ss}$ (17)

E_{cyc} (47)

E_{DEF} (35)

(iii) Indoor air flow rate (SCFM) and external resistance to indoor air flow (inches of water)

(iv) Air temperature (^{o}F)

Outdoor dry bulb
Outdoor wet bulb or dew point
Indoor dry bulb and
Indoor wet bulb

(v) Data as specified in Table II of ASHRAE Standard 37-69

Where the heating capacities $\dot{Q}_{ss}(47)$, $\dot{Q}_{ss}(62)$ and $\dot{Q}_{ss}(17)$ and the indoor air flow rate are calculated using the equations specified in section 3.8.1 and 7.4 of ASHRAE standard 37-69. The total heating done, $Q_{cyc}(47)$ and $Q_{DEF}(35)$ are calculated using the equations below.

Units not having an indoor fan as part of the model tested shall add 1250 Btu/hr per 1,000 SCFM of indoor air handled to the measured capacity to obtain the total heating capacity, $\dot{Q}_{ss}(17)$, $\dot{Q}_{ss}(47)$ or $\dot{Q}_{ss}(62)$, and add 365 watts per 1,000 SCFM of indoor air handled to the measured power to obtain the total power input, $\dot{E}_{ss}(17)$, $\dot{E}_{ss}(47)$, or $\dot{E}_{ss}(62)$, to the unit.

The coefficients of performance (COP) for the High Temperature Tests $COP_{ss}(62)$ or $COP_{ss}(47)$, and Low Temperature

Test, $COP_{ss}(17)$, are calculated as the ratio of the heating capacity in Btu/hr to the product of 3.413 and the power inputs to the indoor fan in watts and the power inputs to the remaining equipment components (including all controls) in watts.

For units which do not have indoor air circulating fans furnished as part of the model shall adjust their total heating done ($Q_{cyc}(47)$) and energy used $E_{cyc}(47)$ in one complete cycle for the effect of circulating indoor air equipment power. For units tested without an indoor fan as part of the model, $Q_{cyc}(47)$ shall be increased by a quantity of heat equal to the product of 1250 Btu/hr per 1,000 SCFM, the length of the on-period of the test cycle in hours, and the flow rate of indoor air circulated in units of 1,000 SCFM. The total energy usage, $E_{cyc}(47)$, shall be the sum of the energy usage required for air circulation during the test cycle and the energy used by the remaining equipment components (including all controls) during the test cycle. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of the on-period of the test cycle in hours, and the rate of indoor air circulated in units of 1000 SCFM.

The cyclic coefficient of performance, $COP_{cyc}(47)$ is calculated as the ratio of the total heating done ($Q_{cyc}(47)$, in Btu's to the product of 3.413 Btu/watt-hour and the total energy usage ($E_{cyc}(47)$) in watt hours.

The net heating capacity, $Q_{DEF}(35)$(Btu/hr), is the total net heating done over the test period (including any credit for the indoor fan heat) divided by the total length of the test period, in hours. For units with "demand defrost control system", the net heating capacity, $Q_{DEF}(35)$, shall be multiplied by an enhancement factor of 1.05 to compensate for improved performance not measured in the Frost Accumulation Test. For units tested without indoor fans, the value determined for $Q_{DEF}(35)$ below shall be increased by a quantity of heat equal to the product of 1250 Btu/hr per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate, of indoor air circulated in units of 1000 SCFM.

The total energy usage, $E_{DEF}(35)$ shall be the sum of the energy usage required for indoor-air circulation during the test period and the energy used by the remaining equipment components during the test period. Units not having an indoor fan as part of the model tested, shall set the energy required for indoor air circulation equal to the quantity given by the product of 365 watts per 1000 SCFM, the length of time in hours during the Frost Accumulation Test that there was indoor air circulating, and the average flow rate of indoor air circulated in units of 1000 SCFM.

The net electrical power to the compressor, $\dot{E}_{DEF}$ (35), is the total energy usage for the test period including the energy to the indoor fan in watt hours divided by the total length of the test period, in hours.

The actual heating done during the Cyclic Test, Q_{cyc} (47), shall be determined using the following equation:

(1) $$Q_{cyc}\ (47) = \frac{(\dot{V} \times \rho \times C_{pa} \times \Gamma)}{(V_{n1} \times (1 + W_n))}$$

where

$\dot{V}$ = the flow rate during the on-period calculated in accordance with section 7.4 of ASHRAE Standard 37-69 in CFM

C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb-°F).

V_{n1} = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).

W_n = Humidity ratio (lb/lb).

and Γ (hr.-°F), which is described by the equation

$$\Gamma = \int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}} [T_{a2}(t) - T_{a1}(t)]\,dt$$

where

$T_{a1}(t)$ = Dry-bulb temperature of air entering the indoor coil (°F) at time (t).

$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

The net heating, Q_{DEF} (35) in Btu's done during the test period shall be determined using the following equation:

(2) $$Q_{DEF}\ (35) = \frac{(\dot{V} \times \rho \times C_{pa} \times \Gamma)}{(V_{n1} \times (1 + W_n))}$$

where

$\dot{V}$ = the average of the air flow rate calculated at four or more time intervals throughout the heating phase of the test using the equation in section 7.4 of ASHRAE Standard 37-69

C_{pa} = Specific heat at constant pressure of air-water mixture per pound of dry air, (Btu/lb-°F).

V_{n1} = Specific volume of air-water mixture at the same dry-bulb temperature, humidity ratio, and pressure used in the determination of the indoor air flow rate (ft³/lb).

W_n = Humidity ratio (lb/lb).

and Γ (hr.-°F), which is described by the equation

$$\Gamma = \int_{\text{(time indoor fan on)}}^{\text{(time indoor fan off)}} [T_{a2}(t) - T_{a1}(t)]\,dt$$

where

$T_{a1}(t)$ = Dry-bulb temperature of air entering the indoor coil (°F) at time (t).

$T_{a2}(t)$ = Dry-bulb temperature of air leaving the indoor coil (°F) at time (t).

(3) cyclic degradation coefficient shall be calculated as follows:

$$C_D = \frac{1 - \frac{COP_{cyc}(47)}{COP_{ss}(47)}}{1 - HLF}$$

where

C_D = the cyclic degradation coefficient rounde to the nearest .01

$COP_{cyc}(47)$ as defined above

$COP_{ss}(47)$ as defined above

HLF is the heating load factor calculated as follows:

$$HLF = \frac{Q_{cyc}(47)}{\dot{Q}_{ss}(47) \times \tau}$$

where

$Q_{cyc}(47)$ as defined above

$\dot{Q}_{ss}(47)$ as defined above

τ = Duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

5.2.1 Calculation of the heating seasonal performance factor (HSPF) for air-source heat pumps with single speed compressors.

For each climatic region listed in section 6.2.4, and for design heating requirements equal to both the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

$$HSPF = \frac{\sum_j \frac{n_j}{N} BL(T_j)}{(3.413)\left[\sum_j \frac{n_j}{N} \frac{X(T_j)}{PLF(X)} \delta(T_j) \dot{E}(T_j) + \sum_j \frac{RH(T_j)}{N}\right]}$$

where

j = 1,2,3, ---, n corresponds to the j^{th} temperature bin

n = total number of non-zero temperature bins in the climatic region

T_j = (67 - 5_j) is the representative temperature of the j^{th} bin, (^oF)

$\sum_j$ = indicated the quantity following the symbol is to be summed over all temparature bins

$\frac{RH(T_j)}{N}$ = supplementary resistance heat term at temperature T_j required in those cases where the heat pump automatically turns off ($T_j < T_{off}$) or when it is needed to meet the balance of the building heating requirements, (kW)

$\frac{n_j}{N}$ = is the number of hours in the j^{th} temperature bin divided by $N \equiv \sum_j n_j$ and is referred to as the "fractional hours in j^{th} temperature bin"

3.413 = is a conversion factor which converts watt hours to Btu

$BL(T_j)$ = building load at temperature T_j, (Btu/hr)

$\delta(T_j)$ = heat pump low temperature cut-out factor

$X(T_j)$ = heat pump heating load factor

PLF(X) = heat pump part load factor

The quantities $BL(T_j)$, $\delta(T_j)$, $X(T_j)$, PLF(X) and $\frac{RH(T_j)}{N}$ are defined by the following equations:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right)(C)(DHR)$$

Where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section 6.26 in kBtu/hr

where

$$(\text{minimum design heating requirement}) = \begin{cases} \dot{Q}_{SS}(47)\,\frac{(65-T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ \dot{Q}_{SS}(47), & \text{for region V} \end{cases}$$

and

$$(\text{maximum design heating requirement}) = \begin{cases} 2\dot{Q}_{SS}(47)\,\frac{(65-T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ 2.2\dot{Q}_{SS}(47), & \text{for region V} \end{cases}$$

where

Q_{SS} (47) is the heat pump capacity measured during the High Temperature Test @ 47^oF

T_{OD} is the outdoor design temperature given in section 6.2.4

$$\delta(T_j) = \begin{cases} 0 \; ; & T_j \leq T_{OFF} \quad \text{or} \quad \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} < 1 \\ \frac{1}{2} \; ; & T_{OFF} < T_j \leq T_{ON} \quad \text{and} \quad \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \geq 1 \\ 1 \; ; & T_j > T_{ON} \quad \text{and} \quad \dfrac{\dot{Q}(T_j)}{(3.413)(\dot{E}(T_j))} \geq 1 \end{cases}$$

$$X(T_j) = \begin{cases} \dfrac{BL(T_j)}{\dot{Q}(T_j)} \; ; & \dot{Q}(T_j) \geq BL(T_j) \\ 1 \; ; & \dot{Q}(T_j) \leq BL(T_j) \end{cases}$$

$$PLF(X) = 1 - C_D(1 - X(T_j))$$

$$\frac{RH(T_j)}{N} = \frac{\left[BL(T_j) - \dot{Q}(T_j)\, X(T_j)\, \delta(T_j)\right] \frac{n_j}{N}}{3.413}$$

where

T_{OFF} = the outdoor temperature that the compressor is automatically shut off at (If no such temperature exists, T_j is always greater than T_{OFF} and T_{ON}.)

T_{ON} = the outdoor temperature that the compressor is automatically turned on (if applicable) if designed for low temperature automatic shut-off.

C_D = degradation factor determined described in section 5.2 and 2.2.1

In using the above equation to calculate HSPF, the heat pump capacity in Btu/hr, $\dot{Q}$, and the power in watts, $\dot{E}$, shall be obtained as follows:

$$\dot{Q}(T_j) = \begin{cases} \dot{Q}_{SS}(17) + \dfrac{(\dot{Q}_{SS}(47) - \dot{Q}_{SS}(17))(T_j - 17)}{30} \; , & T_j \geq 45°F \text{ or } T_j \leq 17°F \\ \dot{Q}_{SS}(17) + \dfrac{(\dot{Q}_{DEF}(35) - \dot{Q}_{SS}(17))(T_j - 17)}{18} \; , & 17°F < T_j < 45°F \end{cases}$$

$$\dot{E}(T_j) = \begin{cases} \dot{E}_{SS}(17) + \dfrac{(\dot{E}_{SS}(47) - \dot{E}_{SS}(17))(T_j - 17)}{30} \; , & T_j \geq 45°F \text{ or } T_j \leq 17°F \\ \dot{E}_{SS}(17) + \dfrac{(\dot{E}_{DEF}(35) - \dot{E}_{SS}(17))(T_j - 17)}{18} \; , & 17°F < T_j < 45°F \end{cases}$$

where $\dot{Q}_{SS}(47)$ and $\dot{E}_{SS}(47)$ and $\dot{Q}_{DEF}(35)$ and $\dot{E}_{SS}(47)$ and $\dot{Q}_{SS}(17)$ and $\dot{E}_{SS}(17)$ are the capacities (in Btu/hr) and powers (in watts) measured during the High Temperature Test, the Frost Accumulation test, and the Low Temperature Test, respectively.

Once the maximum and minimum HSPF and operating cost values have been obtained for each region, the HSPF and operating cost shall be determined for each standardized design heating requirement (see section 6.2.6) between the maximum and minimum design heating requirements by means of interpolation.

5.2.2 Calculation of the heating seasonal performance factor (HSPF) for air source heat pumps with a two-speed compressor, two compressors, or cylinder unloading.

For each climatic region listed in section 6.2.4, and for design heating requirements equal to both the standardized minimum and maximum design heating requirements defined below, calculate the HSPF defined as:

$$HSPF = \frac{\sum_j \frac{n_j}{N} BL(T_j)}{(3.413)\left[\sum_j \frac{E(T_j)}{N} + \sum_j \frac{RH(T_j)}{N}\right]}$$

where

$\sum_j$ as defined in 5.2.1

$\frac{n_j}{N}$ as defined in 5.2.1

3.413 as defined in 5.2.1

T_j as defined in 5.2.1

and $BL(T_j)$ is the building load at temperature T_j, in Btu/hr, calculated by:

$$BL(T_j) = \left(\frac{65 - T_j}{65 - T_{OD}}\right) (C) \ (DHR)$$

where

(C) = 0.77 is a correction factor which tends to improve the agreement between calculated and measured building loads

(DHR) = the minimum and maximum design heating requirement which the heat pump is likely to encounter when installed in a residence, rounded off to the nearest standardized DHR in section 6.2.6 in Btu's per hour

where

$$(\text{minimum design heating requirement}) = \begin{cases} \dot{Q}_{SS}^{(k=2)}(47)\,\frac{(65-\dot{T}_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ \dot{Q}_{SS}^{(k=2)}(47), & \text{for region V} \end{cases}$$

and

$$(\text{maximum design heating requirement}) = \begin{cases} 2\dot{Q}_{SS}^{k=2}(47)\,\frac{(65-T_{OD})}{60}, & \text{for regions I, II, III, IV, and VI} \\ 2.2\,\dot{Q}_{SS}^{k=2}(47) & \text{for region V} \end{cases}$$

where

$\dot{Q}_{ss}^{(k=2)}$ (47) is the heat pump capacity measured during the high temperature performance test at 47°F, with the unit operating at the high compressor speed or with both compressors in operation, in Btu/hr

T_{OD} is the outdoor design temperature given in section 6.2.4 in degrees °F

Note: the superscript (k=1) and (k=2) refer to the heat pump operating at low speed or single compressor operation and high speed or two compressor operation respectively.

$\frac{E(T_j)}{N}$ is the heat pump electrical energy usage in the j^{th} temperature bin divided by the total number of bin hours and is evaluated according to the four possible cases of heat pump operation denoted below in watts

$\frac{RH(T_j)}{N}$ as defined in 5.2.1 and is evaluated according to the four possible cases of heat pump operation denoted below. (in watts)

CASE I

Units operating at low compressor speed or with a single compressor, i.e., k=1, for which the building heating load, $BL(T_j)$ is less than or equal to the heating capacity, $\dot{Q}^{k=1}(T_j)$.

$BL(T_j) \leq \dot{Q}^{k=1}(T_j)$

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=1}(T_j)\, X^{k=1}(T_j)\, \delta'(T_j)}{PLF^{k=1}} \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N} BL(T_j)[1 - \delta'(T_j)]}{3.413}$$

$$X^{k=1}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=1}(T_j)}$$

$$PLF^{k=1} = 1 - C_D^{k=1}(1 - X_j^{k=1})$$

$$\delta'(T_j) = \begin{cases} 0 & ; \quad T_j \leq T_{OFF} \\ \frac{1}{2} & ; \quad T_{OFF} < T_j \leq T_{ON} \\ 1 & ; \quad T_j > T_{ON} \end{cases}$$

CASE II

Units alternating between high speed or two compressor operation (k=2) and low speed or single compressor operation (k=1) to satisfy the building heating load at temperature T_j

$$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = [\dot{E}^{k=1}(T_j)\, X^{k=1}(T_j) + \dot{E}^{k=2}(T_j)\, X^{k=2}(T_j)]\, \delta'(T_j)\, \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N}\, BL(T_j)[1 - \delta'(T_j)]}{3.413}$$

$$X^{k=1}(T_j) = \frac{\dot{Q}^{k=2}(T_j) - BL(T_j)}{\dot{Q}^{k=2}(T_j) - \dot{Q}^{k=1}(T_j)}$$

$$X^{k=2}(T_j) = 1 - X^{k=1}(T_j)$$

$$\delta'(T_j) = \begin{cases} 0 & ; \quad T_j \le T_{OFF} \\ \frac{1}{2} & ; \quad T_{OFF} < T_j \le T_{ON} \\ 1 & ; \quad T_j > T_{ON} \end{cases}$$

CASE III

Units cycling on and off at high compressor speed or cycling both compressors on and off together (k=2) in order to satisfy the building heating load at temperature T_j.

$$\dot{Q}^{k=1}(T_j) < BL(T_j) < \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = \frac{\dot{E}^{k=2}(T_j)\, X^{k=2}(T_j)\, \delta''(T_j)\, \frac{n_j}{N}}{PLF^{k=2}}$$

$$\frac{RH(T_j)}{N} = \frac{\frac{n_j}{N}\, BL(T_j)[1 - \delta''(T_j)]}{3.413}$$

$$X^{k=2}(T_j) = \frac{BL(T_j)}{\dot{Q}^{k=2}(T_j)}$$

$$PLF^{k=2} = 1 - C_D^{k=2}\,(1 - X^{k=2}(T_j))$$

$$\delta''(T_j) = \begin{cases} 0 & ; \quad T_j \le T_{OFF} \\ \frac{1}{2} & ; \quad T_{OFF} < T_j \le T_{ON} \\ 1 & ; \quad T_j > T_{ON} \end{cases}$$

CASE IV

Units operating continuously at high compressor speed or with both compressors in continuous operation (k=2) in order to satisfy the building heating load at temperature T_j.

$$BL(T_j) \ge \dot{Q}^{k=2}(T_j)$$

$$\frac{E(T_j)}{N} = \dot{E}^{k=2}(T_j)\, X^{k=2}(T_j)\, \delta''(T_j)\, \frac{n_j}{N}$$

$$\frac{RH(T_j)}{N} = \frac{[BL(T_j) - \dot{Q}^{k=2}(T_j)\, X^{k=2}(T_j)\, \delta''(T_j)]\, \frac{n_j}{N}}{3.413}$$

$$X^{k=2}(T_j) = 1.0$$

$$\delta''(T_j) = \begin{cases} 0 \ ; & T_j \le T_{OFF} \text{ or } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} < 1 \\ \frac{1}{2} \ ; & T_{OFF} < T_j \le T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \ge 1 \\ 1 \ ; & T_j \ge T_{ON} \text{ and } \dfrac{\dot{Q}^{k=2}(T_j)}{(3.413)(\dot{E}^{k=2}(T_j))} \ge 1 \end{cases}$$

Where in each of the above cases

$X(T_j)$ = heat pump heating load factor

PLF = heat pump part load factor (not required for cases II and III

$\delta(T_j)$ = heat pump low temperature cut-out factor

T_{off} as defined in 5.2.1

T_{on} as defined in 5.2.1

$C_D^{k=2}$ = the part load degradation factor described in section 2.2.1 and 5.2 for the unit cycling at high compressor speed or with both compressors simultaneously cycling (if applicable).

$C_D^{(k=1)}$ = the part load degradation factor described in section 2.2.1 and 5.2 for the unit cycling at low compressor speed or with the single compressor that normally operates at low heating loads.

$$\dot{Q}^{k=1}(T_j) = \begin{cases} \dot{Q}^{k=1}_{SS}(47) + \dfrac{\left(\dot{Q}^{k=1}_{SS}(62) - \dot{Q}^{k=1}_{SS}(47)\right)(T_j - 47)}{15} \ ; & T_j \ge 40°F \\ \dot{Q}^{k=1}_{SS}(17) + \dfrac{\left(\dot{Q}^{k=1}_{DEF}(35) - \dot{Q}^{k=1}_{SS}(17)\right)(T_j - 17)}{18} \ ; & 17°F \le T_j < 40°F \\ \dot{Q}^{k=1}_{SS}(17) + \dfrac{\left(\dot{Q}^{k=1}_{SS}(47) - \dot{Q}^{k=1}_{SS}(17)\right)(T_j - 17)}{30} \ ; & T_j < 17°F \end{cases}$$

$$\dot{Q}^{k=2}(T_j) = \begin{cases} \dot{Q}^{k=2}_{SS}(17) + \dfrac{\left(\dot{Q}^{k=2}_{SS}(47) - \dot{Q}^{k=2}_{SS}(17)\right)(T_j - 17)}{30} \ ; & T_j \ge 45°F \text{ or } T_j \le 17°F \\ \dot{Q}^{k=2}_{SS}(17) + \dfrac{\left(\dot{Q}^{k=2}_{DEF}(35) - \dot{Q}^{k=2}_{SS}(17)\right)(T_j - 17)}{18} \ ; & 17°F < T_j < 45°F \end{cases}$$

$$\dot{E}^{k=1}(T_j) = \begin{cases} \dot{E}^{k=1}_{SS}(47) + \dfrac{\left(\dot{E}^{k=1}_{SS}(62) - \dot{E}^{k=1}_{SS}(47)\right)(T_j - 47)}{15} \ ; & T_j \ge 40°F \\ \dot{E}^{k=1}_{SS}(17) + \dfrac{\left(\dot{E}^{k=1}_{DEF}(35) - \dot{E}^{k=1}_{SS}(17)\right)(T_j - 17)}{18} \ ; & 17°F \le T_j < 40°F \\ \dot{E}^{k=1}_{SS}(17) + \dfrac{\left(\dot{E}^{k=1}_{SS}(47) - \dot{E}^{k=1}_{SS}(17)\right)(T_j - 17)}{30} \ ; & T_j < 17°F \end{cases}$$

$$\dot{E}^{k=2}(T_j) = \begin{cases} \dot{E}^{k=2}_{SS}(17) + \dfrac{\left(\dot{E}^{k=2}_{SS}(47) - \dot{E}^{k=2}_{SS}(17)\right)(T_j - 17)}{30} \ ; & T_j \ge 45°F \text{ or } T_j \le 17°F \\ \dot{E}^{k=2}_{SS}(17) + \dfrac{\left(\dot{E}^{k=2}_{DEF}(35) - \dot{E}^{k=2}_{SS}(17)\right)(T_j - 17)}{18} \ ; & 17°F < T_j < 45°F \end{cases}$$

Calculate the heating seasonal performance factors and seasonal operating costs corresponding to the standardized maximum and minimum design heating requirements for each of the six regions specified in Section 6.2.5. When the maximum and minimum HSPF's and operating cost values have been obtained for each region, the HSPF, and operating cost shall be determined for each standardized design heating requirement (see section 6.2.6) between the maximum and minimum design heating requirements of that particular unit by means of interpolation.

5.3 Calculations of the Actual Representative Regional Annual Performance Factors for Air Source Central Air Conditioners (Heat Pumps) Which Provide Both Heating and Cooling.

5.3.1 Calculation of actual regional annual performance factors (APF_A) for a particular location and for each standardized design heating requirement.

$$(APF_A) = \frac{(CLH_A)\ (\dot{Q}_{ss}^{k}(95F))\ +\ (HLH_A)\ (DHR)\ (C)}{\frac{(CLH_A)\ (\dot{Q}_{ss}^{k}(95F))}{(CSPF)}\ +\ \frac{(HLH_A)\ (DHR)\ (C)}{(HSPF)}}$$

where

(CLH_A) is the actual cooling load hours for the particular location, determined from the map in section 6.1.3

$(\dot{Q}_{ss}^{k}(95F))$ is defined in 5.1

(DHR) is defined in 5.2.2

(C) is defined in 5.2.2

(HLH_A) is the actual heating load hours for the particular location determined from the map in section 6.2.5

(CSPF) is the cooling seasonal performance factor as determined in section 5.1

(HSPF) is the heating seasonal performance factors as determined in section 5.2 for each standardized design heating requirement within the particular location's region or for the actual design heating requirement if known.

where

the particular location's region is determined from the map in section 6.2.5 and, the standardized design heating requirements within the region are determined in sections 5.2 and 6.2.6

5.3.2 Calculation of representative regional annual performance factors (APF_R) for each region and for each standardized design heating requirement

$$(APF_R) = \frac{(CLH_R)\ (\dot{Q}_{ss}^{k}(95F))\ +\ (HLH_R)\ (DHR)\ (C)}{\frac{(CLH_R)\ (\dot{Q}_{ss}^{k}(95F))}{CSPF}\ +\ \frac{(HLH_R)\ (DHR)\ (C)}{HSPF}}$$

$(\dot{Q}_{ss}^{k}(95F))$ is defined in 5.1

(DHR) is defined in 5.2.2

(C) is defined in 5.2.2

where

(CLH_R) is the representative cooling load hours for each heating load hours region, as determined in section 6.3.

(HLH_R) is the representative heating load hours for each region as determined in section 6.2.5

(CSPF) is the cooling seasonal performance factor as determined in section 5.1

(HSPF) is the heating seasonal performance factor as determined in section 5.2 for each region and for each standardized design heating requirement within each region.

where

the regions are listed in section 6.2.5 and, the standardized design heating requirements within the regions are determined in sections 5.2 and 6.2.6

5.4 Calculations of Cooling Seasonal Performance Factor for Water Source Units Which Provide Both Heating and Cooling. The cooling seasonal performance factor (CSPF) for water-source heat pumps shall be calculated from the following equation:

$$CSPF_j = \frac{(T_j - T_F)\ COP_E - COP_F) + COP_F}{(T_E - T_F)}$$

j = any location from 6.4 of this Appendix

Where T_j = ground water temperature of a given location in (^oF) from 6.3 of this Appendix or a known design inlet water temperature between 50^oF and 90^oF.

T_E = the inlet water temperature (^oF) as described in test E from 3.4.1 and 3.4.3 of this appendix

T_F = the inlet water temperature (^oF) as described in test F from 3.4.1

$$COP_E = \frac{\dot{Q}_E}{\dot{E}_E\ (3.413)}$$

Where

COP_E is the coefficient of performance from test E detailed in sections 2.4, 3.4, and 4.4 of this Appendix

$\dot{Q}_E$ is the steady state cooling capacity (Btu/hr) obtained from test E described above

$\dot{E}_E$ is the electric input (watts) obtained from test E described above, which shall include a factor of 60 watts per 12,000 Btu/hr of capacity measured in test F detailed in section 2.4, 3.4, and 4.4 of this Appendix to account for pump usage required to move the water through the heat exchanger

(3.413) is the conversion factor for Btu's per watt-hour

$$COP_F = \frac{\dot{Q}_F}{\dot{E}_F \ (3.413)}$$

Where

COP_F is the coefficient of performance from test B detailed in sections 2.4, 3.4, and 4.4 of this Appendix

$\dot{Q}_F$ is the steady state cooling capacity (Btu/hr) obtained from test F described above

$\dot{E}_F$ is the electrical input (watts) obtained from test B described above, which shall include a factor of 60 watts per 12,000 Btu/hr of capacity measured in test F described above, to account for pump usage required to move the water through the heat exchanger.

(3.413) is the conversion factor for Btu's per watt-hour.

5.5 Calculation of Heating Seasonal Performance Factor for Water Source Units Which Provide Both Heating and Cooling. The heating seasonal performance factor (HSPF) for water-source heat pumps shall be calculated from the following equation:

$$HSPF_j = \frac{(T_j - T_H)}{(T_G - T_H)} \ (COP_G - COP_H) + COP_H$$

Where

j = any location from 6.4 of this Appendix

T_j = as described in 5.4 of this Appendix

T_G = the inlet water temperature (^oF) as described in test G from 3.4.2 and 3.4.4 of this Appendix.

T_H = the inlet water temperature (^oF) as described in test H from 3.4.2 and 3.4.4 of this Appendix.

$$COP_G = \frac{\dot{Q}_G}{\dot{E}_G \ (3.413)}$$

Where

COP_G is the coefficient of performance from test G detailed in section 2.4, 3.4, and 4.4 of this Appendix.

$\dot{Q}_G$ is the steady state heating capacity (Btu/hr) obtained from test G as described above.

$\dot{E}_G$ is the electric input (watts) obtained from test G as described above, which shall include a factor of 60 watts per 12,000 Btu/hr of capacity measured in test E as described above, to account for pump usage required to move the water through the heat exchanger.

(3.413) is the conversion factor for Btu per watt-hour.

$$COP_H = \frac{\dot{Q}_H}{\dot{E}_H \ (3.413)}$$

Where

COP_H is the coefficient of performance from test H detailed in sections 2.4, 3.4, and 4.4 of this Appendix

$\dot{Q}_H$ is the steady state heating capacity (Btu/hr) obtained from test G as described above.

$\dot{E}_H$ is the electric input (watts) obtained from test H as described above, which shall include a factor of 60 watts per 12,000 Btu/hr of capacity measured in test G as described above, to account for pump usage required to move the water through the heat exchanger.

(3.413) is the conversion factor for Btu per watt-hour.

5.6 Calculation of Annual Performance Factor For Water Source Units Which Provide Both Heating and Cooling. The annual performance factor (APF) for water-source heat pumps shall be calculated from the following equation:

$$APF_j = \frac{(HSPF_j \times HLH_j) + (CSPF_j \times CLH_j)}{HLH_j + CLH_j}$$

Where

j = any location from 6.4 of this Appendix

$HSPF_j$ = heating seasonal performance factor for any location j as calculated in 5.5 of this Appendix

$CSPF_j$ = heating seasonal performance factor for any location j as calculated in 5.4 of this Appendix

HLH_j = heating load hours for any location j from 6.2.5 of this Appendix.

CLH_j = cooling load hours for any location j from 6.1.3 of this Appendix.

6.0 Reference Material

6.1 Cooling reference material

6.1.1 —*Test operating and test condition tolerance for cyclic dry-coil tests*

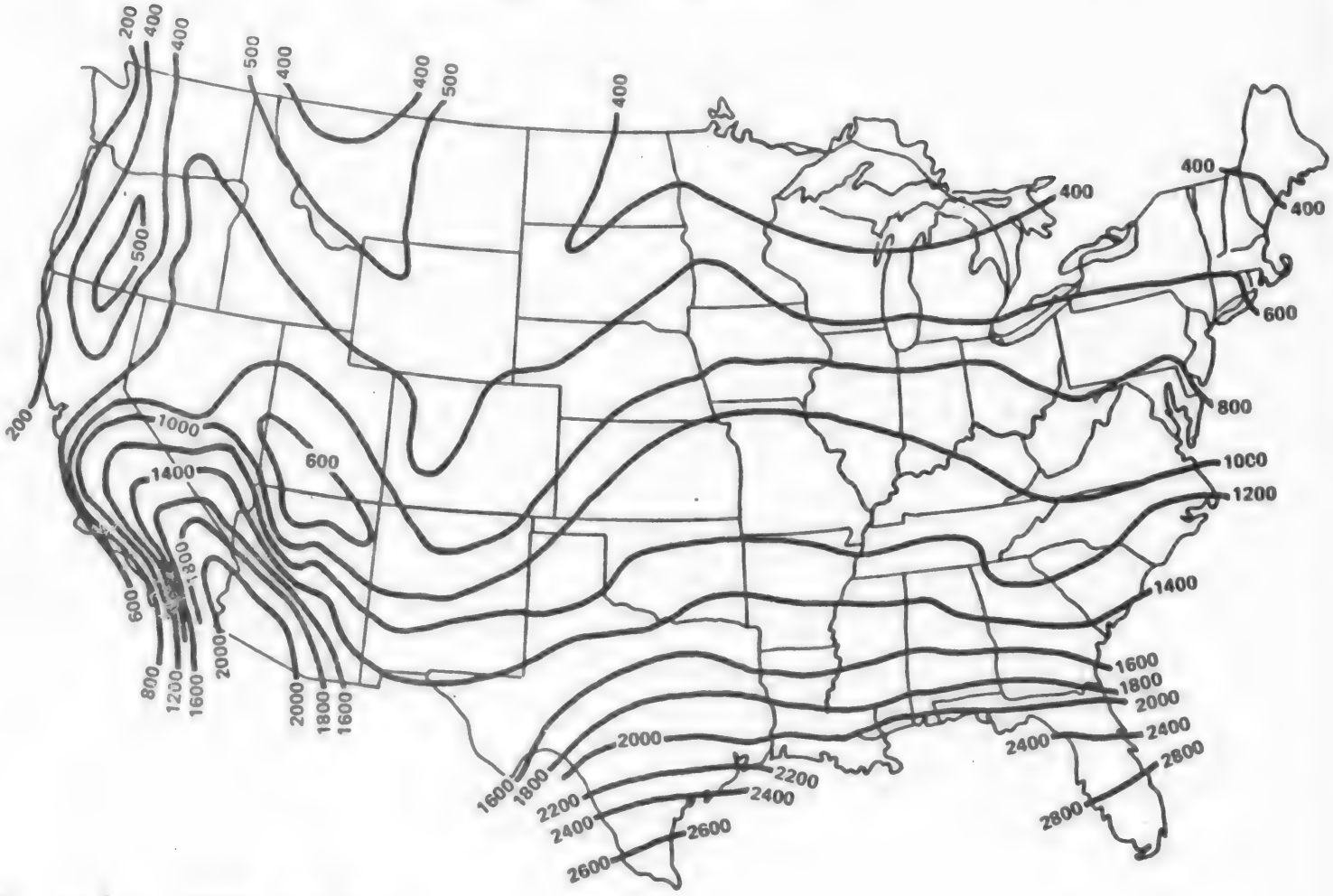
Readings, remarks	Test operating tolerance (total observed range)	Test condition tolerance (variation of average from specified test condition)
Outdoor dry-bulb air temperature, Fahrenheit: Entering	2.0	0.5
Indoor dry-bulb air temperature, Fahrenheit: Entering	2.0	.5
Indoor wet-bulb air temperature, Fahrenheit: Entering	(1)	(1)
After the 1st 30 s after compressor startup.		
External resistance to airflow, inches water	.05	.02
Nozzle pressure drops, percent of reading	2.0	
Electrical voltage inputs to the test unit, percent	2.0	

[1] Shall at no time exceed that value of the wet-bulb temperature which results in the production of condensate by the indoor coil at the dry bulb temperature existing for the air entering the indoor portion of the unit.

6.1.2 —*Distribution of fractional hours in temperature bins to be used for calculation of the SEER for 2-speed compressor and 2-compressor units*

	Bin temperature range (degrees Fahrenheit)	Representative temperature bin for (degrees Fahrenheit)	Fraction of total temperature bin hours n/N
Bin No., j:			
1	65 to 69	67	.214
2	70 to 74	72	.231
3	75 to 79	77	.216
4	80 to 84	82	.161
5	85 to 89	87	.104
6	90 to 94	92	.052
7	95 to 99	97	.018
8	100 to 104	102	.004

6.1.3 Distribution of Actual Cooling Load Hours (CLH_A) Throughout the United States



Alaska	– 0	Cooling	Load	Hours	
Canal Zone	– 6,000	"	"	"	
Guam	– 6,600	"	"	"	
Hawaii	– 2,300	"	"	"	
Puerto Rico	– 6,000	"	"	"	
Samoa	– 6,600	"	"	"	
Virgin Island	– 6,000	"	"	"	

6.2 Heating Reference Material

6.2.1 Test operating and test condition tolerance for

Steady-State High Temperature Test [at 47°F (8.3°C) or 62°F (16.7°C)]

and

Low Temperature Test [at 17°F (-8.3°C)]

	Test Operating* Tolerance	Test Condition** Tolerance
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	---
Indoor wet-bulb, °F		
Entering	1.0	---
Leaving	1.0	---
Outdoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	2.0	---
Outdoor wet-bulb, °F		
Entering	1.0	0.3
Leaving	1.0	---
External resistance to air flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

6.2.2 Test operating and test condition tolerances for the on-period portion of cyclic performance tests.

	Test Operating Tolerances* (Applies after the 1st 30 sec. after compressor start-up)	Test Condition Tolerance**
Indoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	---	---
Indoor wet-bulb, °F		
Entering	1.0	---
Leaving	---	---
Outdoor dry-bulb, °F		
Entering	2.0	0.5
Leaving	---	---
Outdoor wet-bulb, °F		
Entering	2.0	1.0
Leaving	---	---
External resistance to air-flow, inches of water	.05	.02
Electrical voltage, %	2.0	---

* Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value.

** Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition.

6.2.3 Test operating and test condition tolerances for frost accumulation tests

	Testing Operating Tolerance[1] During Heating	Testing Operating Tolerance[1] During Defrost	Test Condition Tolerance[2] (Heating Portion Only)
Indoor dry-bulb, °F			
Entering	2.0	4.0[3]	0.5
Leaving	---	---	---
Indoor wet-bulb, °F			
Entering	1.0	---	---
Leaving	---	---	---
Outdoor dry-bulb, °F			
Entering	2.0	10.0	1.0
Leaving	---	---	---
Outdoor dew-point, °F			
Entering	1.5	---	0.7
Leaving	---	---	---
External resistance to air-flow, inches of water	.05	---	.02
Electrical voltage, %	2.0	---	---

1 Test Operating Tolerance is the maximum permissable variation of any measurement. When expressed as a percentage, the maximum allowable variation is the specified percentage of the average value. Test Operating Tolerance During Heating applies when the heat pump is in the heating mode, except for the first 5 minutes after termination of a defrost cycle. Test Operating Tolerance During Defrost applies during a defrost cycle and during the first 5 minutes after the termination of a defrost cycle when the heat pump is operating in the heating mode.

2 Test Condition Tolerance is the maximum permissable variation of the average value of the measurement from the standard or desired test condition. Test Condition Tolerance applies only when the heat pump is operating in the heating mode.

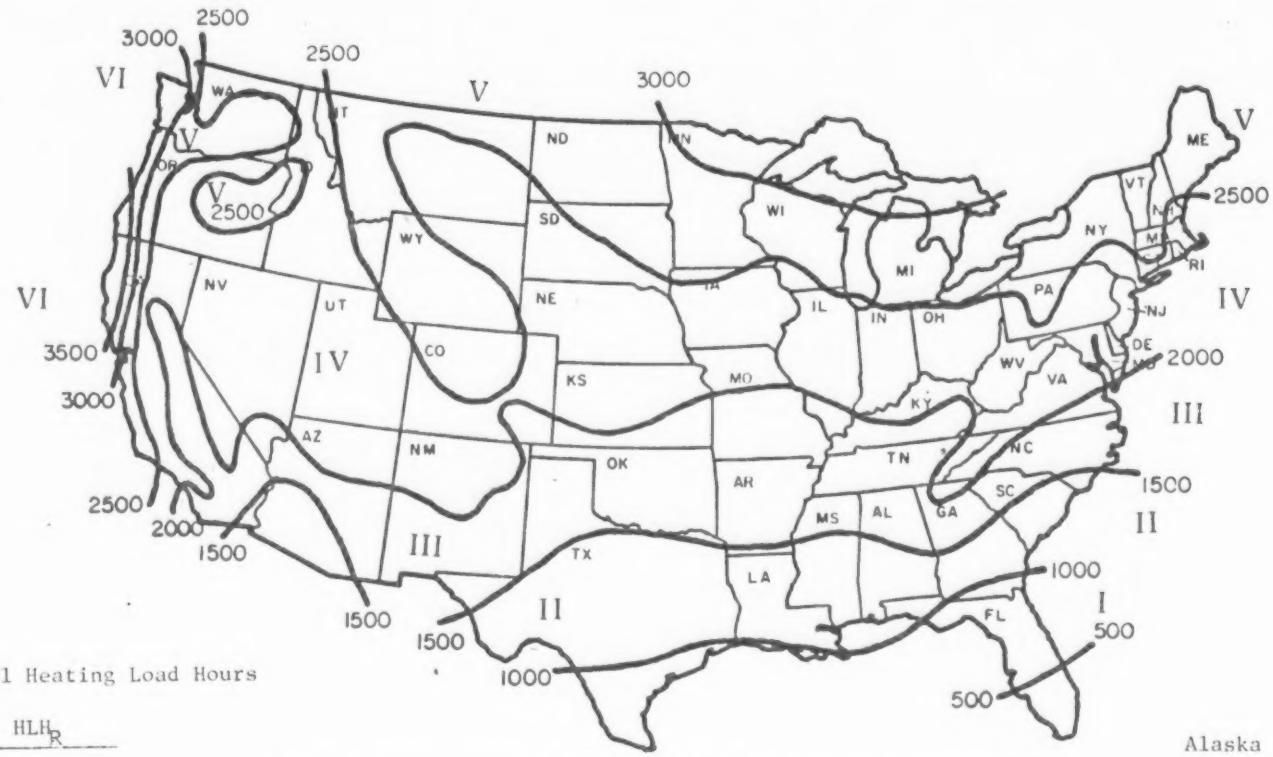
3 Not applicable during defrost if the indoor fan is off.

6.2.4 Distribution of fractional hours in temperature bins, heating load hours and outdoor design temperature for the different climatic regions

Region		I	II	III	IV	V	VI
Heating Load Hours, HLH		750	1250	1750	2250	2750	2750*
Outdoor Design Temperature, T_{OD} for the region		37	27	17	5	-10	30
Fractional Hours:							
Bin #	T_j (°F)						
j = 1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	22	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

* In Pacific Coast Region

6.2.5 Actual heating load hours (HLH_A) and regional heating load hours (HLH_R) for the United States



Regional Heating Load Hours

Region	HLH_R
I	750
II	1250
III	1750
IV	2250
V	2750
VI	2750

This map is reasonably accurate for most parts of the United States but is necessarily highly generalized, and consequently not too accurate in mountainous regions, particularly in the Rockies

Alaska - 3500 HLH
Hawaii and Territories - 0 HLH

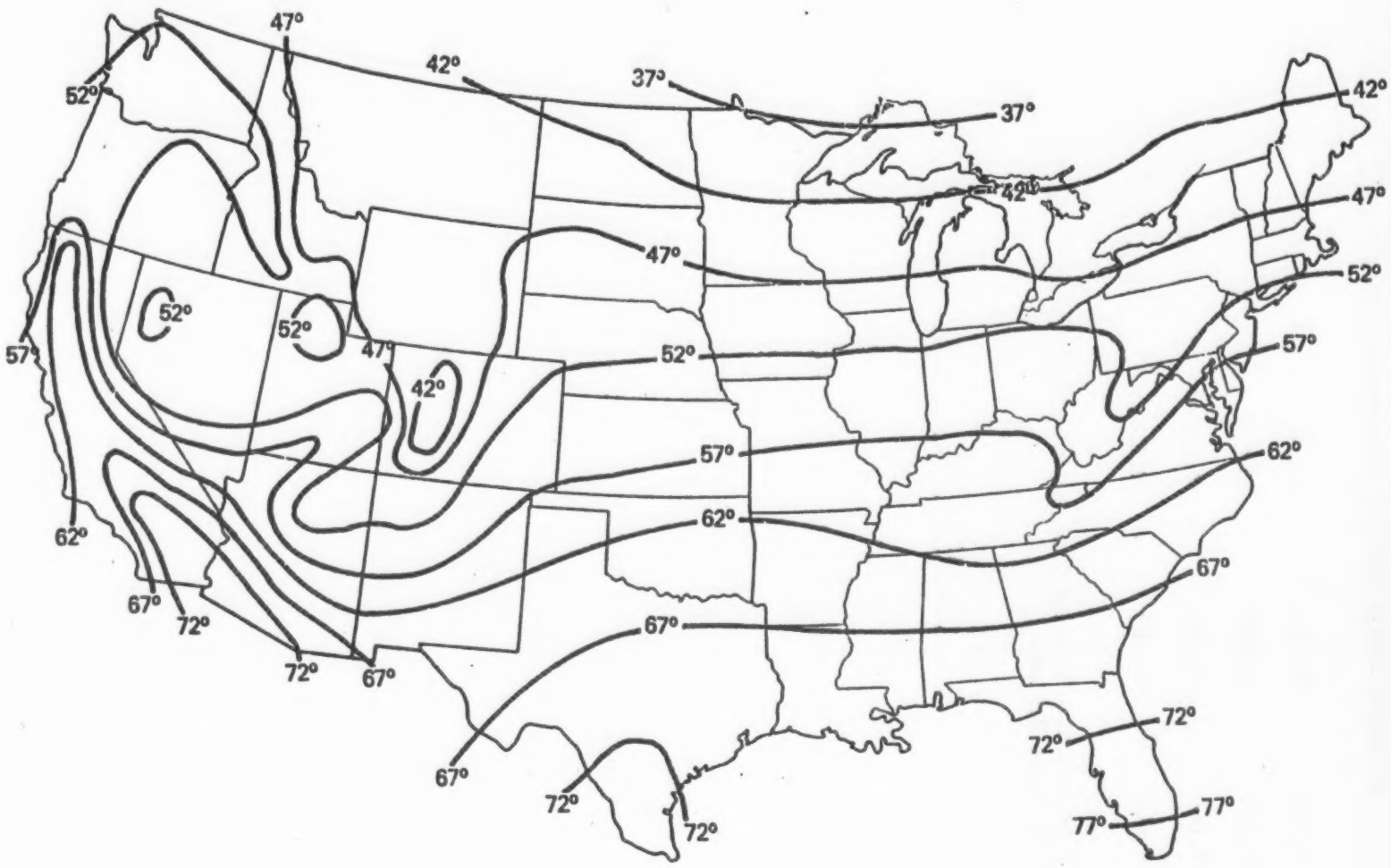
6.2.6 Standard Design Heating Requirements (Btu/hr)

5,000	25,000	50,000	90,000
10,000	30,000	60,000	100,000
15,000	35,000	70,000	110,000
20,000	40,000	80,000	130,000

6.3 Representative Cooling Load Hours (CLH_R) for Each Heating Load Hours Region

Region	CLH_R	HLH_R
I	2400	750
II	1800	1250
III	1200	1750
IV	800	2250
V	400	2750
VI	200	2750

6.4 Ground Water Temperature Map



AVERAGE TEMPERATURE OF SHALLOW GROUND WATER IN DEGREES FAHRENHEIT